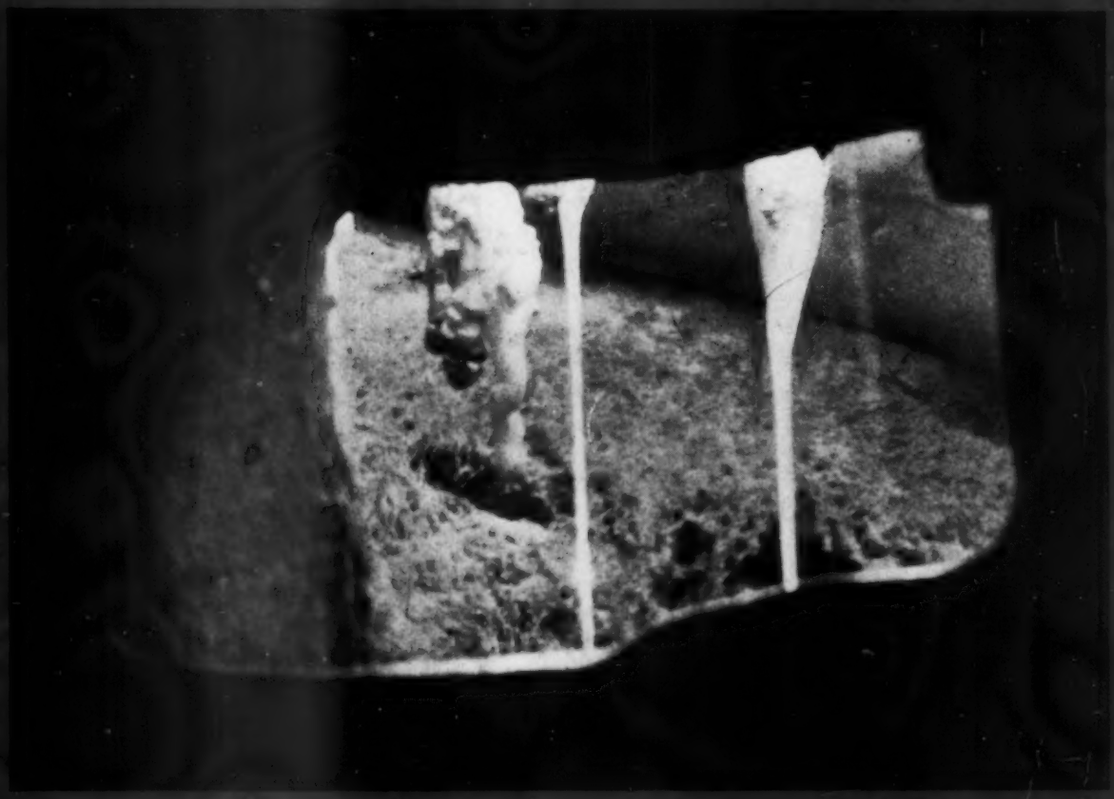
COMBUSTION

DEVOTED TO THE ADVANCEMENT OF STEAM PLANT DESIGN AND OPERATION

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Looking into side of slag-drip chamber in one of the units at Waterside Station, New York; see page 18

Slag-Bottom Furnaces

The Selection of Economizers and Air Preheaters

Design of High-Capacity Boilers

A correction

of the advertisement reproduced below which appeared in June "Combustion" and June 4th "Electrical World"

How Big Should a Pulverized Fuel Furnace Be?

The obvious answer to this question is.

"big enough to permit practical completion of the combustion process in a minimum amount of space and to provide sufficient heat absorption to reduce gas temperature to a point which will prevent excessive slagging of boiler tubes."

The space required to reduce the gas temperature to the point necessary for satisfactory operation depends upon the amount of heat-absorbing surface enclosing the space and the efficiency of that surface. It is a fact that *bare tube* water wall surface is the most efficient practical medium for reduction of furnace gas temperature. This has long been the characteristic construction of C-E Pulverized Fuel Furnaces.

The space required to complete combustion is determined by the rate of combustion. Because the term "combustion rate" has been commonly used to designate the amount of heat released in a furnace, in terms of Btu per hr per cu ft of total furnace volume, it is important to realize that this term, so used, does not in any sense indicate the true rate of combustion. The distinction is especially important at this time in view of the emphasis that has been placed of late on obtaining so-called higher combustion rates in order to permit the use of smaller and presumably less costly furnaces. Accordingly, we present the following discussion of this subject based upon a recent report of our Research Department.

THE combustion rate of a pulverized fuel furnace is commonly expressed as the heat delivered to or liberated in the furnace per cubic foot of furnace volume per hour. This is a nominal rather than the actual rate of combustion.

The combustion of a solid fuel is not subject to an exact mathematical analysis since the concentration of combustible and oxygen cannot be expressed on the same basis. However, with adequate mixing at a temperature above the ignition point, the rate of burning must be proportional to the product of the concentrations of the reacting substances. The actual rate at which fuel burns in a furnace must vary greatly along the path of the gases. The oxygen concentration at the start of combustion is that of the atmosphere, 21 per cent by volume. The combustible content of the suspended coal at this point is about 90 per cent. As the gases leave the furnace, the oxygen content of the furnace atmosphere has been reduced to about 2 per cent by volume and the unburned combustible is less than 2 per cent. At these relative concentrations, the rate of burning in the burner zone must be about 500 times the rate at the furnace outlet.

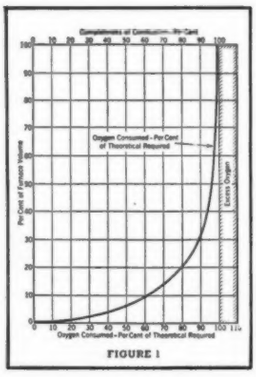


FIGURE 1

The rate of combustion of fuel may be determined by the rate of disappearance of free oxygen. For a furnace designed to burn fuel to an economic degree of completion, say 98 per cent or better, the disappearance of free oxygen would be as shown by the accompanying curve, Fig. 1. The ordinate is divided into per cent of furnace volume along the path of the flame. While the gas velocity through the furnace will decrease as the temperature is reduced, these divisions may also be considered to represent equal intervals of time.

The abcissa shows the reduction in the per cent of free oxygen in the furnace atmosphere incident to the progress of combustion. Since the divisions of the ordinate represent approximately equal time intervals, the slope of the curve at any point represents the time rate of disappearance of oxygen and therefore the time rate of combustion at that point.

Let us consider, for example, a furnace 25 feet square in plan and 30 feet high of a design as shown in Fig. 2, which represents a corner-fired pulverized coal furnace with a continuous slagging bottom. The nominal combustion rate will be, say 35,000 Btu per cubic foot per hour. The hearth will be at a temperature of 2800 F. and the gases as they leave the furnace about 2000 F.

On the basis of the oxygen disappearance curve, Fig. 1, it may be seen that the combustion is 96 per cent complete in the first 50 per cent of the furnace volume. *The nominal combustion rate of the first half of the furnace is therefore 35,000 times .96 divided by .50 or 67,000 Btu per cubic foot per hour. The combustion rate for the first 25 per cent of the furnace will be 35,000 times .85 divided by .25 or 119,000 Btu.* Similarly, the combustion may be computed for any portion of the furnace.

As indicated, these are nominal rates of combustion. By integrating the curve and thus obtaining the summation of the products of small increments of combustible and the rate at which each increment is burned, the true rate of combustion would be determined. *For the furnace shown, this true or weighted average rate of combustion would be 104,000 Btu per hour per cubic foot of volume for the entire furnace.*

If we should now increase the height of the furnace, the volume will be correspondingly increased and the nominal rate of combustion will be lowered. Thus, if we add 5 feet to the height, the nominal combustion rate for the entire furnace will be reduced from 35,000 to 30,000 Btu per cubic foot per hour, the gases leaving the furnace will be reduced about 200 degrees F. in temperature and the amount of unburned carbon slightly reduced. The actual rate of burning below the 30-ft. level will, however, remain unchanged.

To carry this reasoning to the extreme, let us increase the furnace height to infinity. The nominal combustion rate will now be zero, yet the actual rate of burning in the first 30 feet will be the same and we will still have a temperature of 2800 F. at the furnace hearth.

It will be seen from the foregoing that the furnace combustion rate as commonly expressed has nothing at all to do with the actual rate at which the fuel is burned. It is only a convenient method of visualizing the size of a furnace.

The modern water-cooled furnace has a double function. It must be proportioned so that the fuel will be economically burned and it must provide radiant heat-absorbing surface so that the gases will be sufficiently reduced in temperature to avoid excessive slagging of the boiler convection surface. These two processes, heat evolution and heat absorption, take place simultaneously. While the predominant duty of the first part of the furnace is heat evolution and the predominant duty of the latter part heat absorption, there is no place where the furnace may be logically divided on the basis of these functions. Combustion is essentially a continuous process. The latter part of the furnace is no less a furnace by reason of its lower rate of combustion.

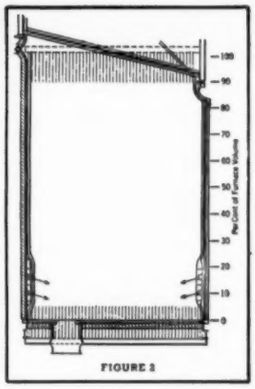


FIGURE 2

No present-day method of pulverized coal firing achieves higher actual rates of combustion than C-E corner firing. No present type of furnace wall construction achieves higher rates of heat absorption per square foot than the C-E *Bare Tube* Water Cooled Wall. By reason of these two facts, C-E Corner Fired Furnaces can be and are designed to provide the *minimum* volume required for satisfactory performance. Furthermore, their design recognizes the fundamental that combustion is essentially a continuous process and accordingly provides for the simultaneous evolution and absorption of heat in the simplest, most efficient and most practical manner possible.

Any saving in first cost of a high-capacity steam generating unit that could be obtained by reducing furnace volume below the minimum required to insure attainment of design performance in conjunction with high availability is certain to be a penny-wise, pound-foolish policy. The cost of maintaining a unit so designed in operable condition, plus the losses chargeable to outage, would quickly dissipate any likely initial saving.

C-E Corner Fired Furnaces of both slagging and dry-bottom types have demonstrated in many plants their ability to meet the highest performance standards achieved in modern practice. Especially is this true with respect to availability—the most significant criterion of sound design and good construction. The acceptance C-E corner firing has achieved is evidenced by the fact that 64 corner-fired units have been purchased to date in this country and an even larger number have been installed in foreign countries. The aggregate rated capacity of such units in this country is 20,000,000 lb of steam per hr.

COMBUSTION ENGINEERING COMPANY, Inc.

200 Madison Avenue, New York, N. Y. *Canadian Associates, Combustion Engineering Corporation, Ltd., Montreal*

ALL TYPES OF BOILERS, STOKERS, PULVERIZED FUEL SYSTEMS AND HEAT RECOVERY EQUIPMENT

An inadvertent error in the above advertisement has been brought to our attention. The figure encircled should be "100" instead of "200". Corrected, the sentence would read, "Thus, if we add 5 feet to the height, the nominal combustion rate for the entire furnace will be reduced from 35,000 to 30,000 Btu per cubic foot per hour, the gases leaving the furnace will be reduced about 100 degrees F. in temperature and the amount of unburned carbon slightly reduced."

The amount of reduction of gas temperature which would be effected as a result of the assumption made in this sentence is essentially an approximation since it depends on a number of factors which would vary greatly in different installations. In fact, some of these factors, such as fuel characteristics, cleanliness of heating surfaces, etc., might vary considerably over a period of time in the same unit. However, for the conditions given and assuming clean wall surfaces, the reduction in gas temperature would be in the neighborhood of 100 F.

Although this figure is not material to the question of combustion rates in modern pulverized fuel furnaces, we are publishing this correction for the benefit of those who may have clipped the advertisement for their reference files on the subject.

Corrected reprints of the advertisement will be sent on request to those interested. Address your letter for the attention of the Publicity Department and ask for extra copies if you can make use of them.

COMBUSTION ENGINEERING COMPANY, Inc.

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VOLUME TEN NUMBER ONE

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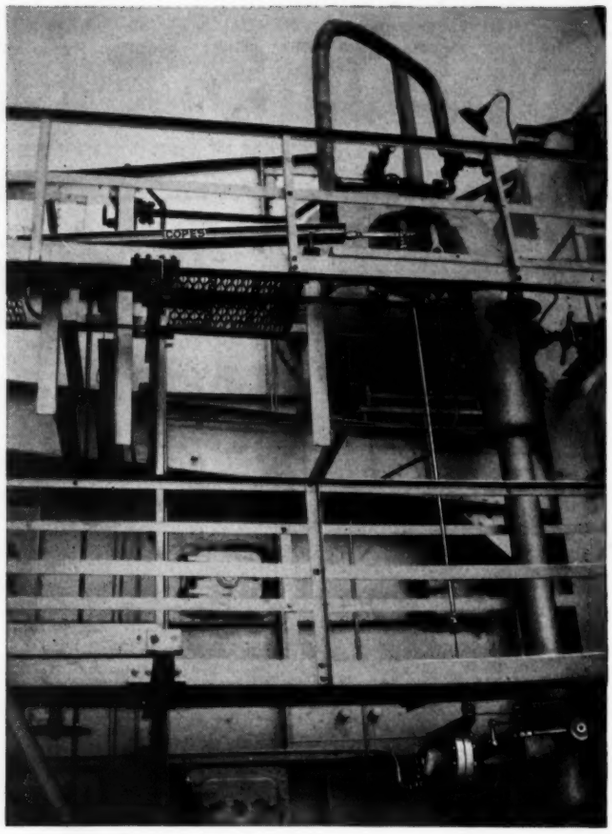
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EDITORIAL

Capacity Margins

Engineering is defined by the dictionary as "The art and science by which the mechanical properties of matter are made useful to man in structures and machines." While the properties of matter are now well known, engineering in certain fields can never be an exact science, such as is astronomy, for instance, because of the numerous variable and dependent factors entering into their application. Nevertheless, research and experience have combined to remove much of the uncertainty from the calculations of the designing engineer, especially with reference to predicted performance.

There was a time, not so long ago, when designers of power plant equipment, such as boilers, prime movers and certain auxiliaries, viewed with keen satisfaction the attainment of vast overload capacities or efficiencies far in excess of those specified. The practice became so general that purchasers came to regard the accumulated excess capacity of several units as providing the desired reserve—a condition not possible in a plant laid out on the unit principle. The responsibility, of course, lay with those designers who were accustomed to "play safe" in the face of rapidly advancing practice and the unavailability of certain basic data.

This, in most cases, is now neither necessary nor permissible in view of accumulated knowledge and the stringent requirements to be met. The advent of high steam temperatures has, more than any other factor, been responsible for requiring the exact proportioning of steam generating units. Hence, present designs approach the objectives more closely and leave much less margin than heretofore.

Therefore, under highly competitive conditions, such as now exist in most lines of power plant equipment, the purchaser may rightfully expect that which he specifies and pays for, together with reliability and satisfactory performance, but it is unreasonable for him to expect any considerable excess.

Slag Disposal

While both the dry-bottom and the slagging furnaces have their respective places, depending upon the coal, load characteristics and size of unit, it is significant that fully half of the more recent high-capacity units employ the latter. These are practically all of the continuous-drip type in contrast with the earlier intermittent, or slag-tap, type. The advantages of the continuous type are discussed by Mr. Caldwell in this issue, but some further observations on ash disposal may be in order.

Most large power stations have their ash disposal problems, the solutions of which depend to a considerable extent on plant location. In the days of active building construction the demand for stoker ash in many sections was sufficient to more than pay for its handling, but in the case of pulverized coal installations the greater part of the fine ash was discharged up the stack to become diffused in the atmosphere and deposited over a wide area. As municipalities became more pure-air conscious the application of dust recovery equipment was extended, but the large quantity of such ash to be handled often greatly increased the installation cost. However, the introduction of the slagging furnace, which entraps a large portion of the fine ash, has resulted not only in a reduction in the size and consequent cost of the recovery equipment required, but also in the amount of such ash requiring ultimate disposal.

Many studies have been made with a view to finding commercial uses for fly ash and some degree of success has been attained, particularly in the making of building blocks although the difficulty has been to find applications that would utilize the large bulk available.

As to the disposal of quenched slag from a slagging furnace, some of the earlier applications were in plants so situated as to permit sluicing it to fill in adjacent property, but few such opportunities remain and in the majority of cases it is removed by barges, railway cars or trucks. Its handling, however, offers less obstacles than that of the fine ash.

Furnace slag has also been the subject of considerable study in an effort to discover commercial uses that would compensate for its handling. One such application has been railway fill and ballast for which it has been used to some extent, but here again, under present conditions, the demand is small. More recently, however, there has been developed another outlet which appears promising. This is in connection with the manufacture of steel where it serves as a fluxing agent and, because of its high iron content, saves in the use of high-grade ore. Already in one locality the slag from a large central station is being utilized in this manner and the return is reported as paying for the delivery and handling costs.

Power Allocations

In his presidential address before the Edison Electric Institute at Atlantic City last month, C. W. Kellogg cited figures showing that the electric utility industry, during the last ten years, had expended approximately five and a half billion dollars on construction and had brought electricity to seven million new customers. When it is considered that nearly half this period covered the depression years, it is reasonable to expect that the next decade should offer still greater opportunities, if certain unfair phases of governmental competition, in the form of subsidies and seemingly improper allocation of costs on Federal Projects, could be rectified. In reference to the latter, Mr. Kellogg pointed to the 22 per cent of the total cost of the Columbia River Development and 37½ per cent of the TVA development as having been allocated to power generation. This is despite the fact that, in the case of TVA, power appears to be the "dog" and flood control the "tail."

SLAG-BOTTOM FURNACES



By W. E. CALDWELL
Consolidated Edison Co. of New York, Inc.

The merits of the continuous slagging-bottom furnace are reviewed with reference to its stabilizing effect on combustion, ability to handle non-premium coals, entrainment of fine ash, ease of ash disposal and low capital costs. Comparison is made with the intermittent slag-tap type, and various arrangements of slag opening, methods of cooling and bottom coverings are discussed. Typical views in such a furnace under load are included.

Fig. 1—Interior of furnace when cold, looking toward one of the corner burners and showing slag-drip opening in floor

ALTHOUGH the pulverized coal slagging furnace is a comparatively recent development, it has found wide acceptance due to certain attractive advantages. In the choice of steam generating equipment consideration of sound economics usually dictates a design which will produce steam at the lowest cost consistent with load, capital, labor and fuel conditions. These conditions vary widely and it is not likely that any single type of combustion system or ash-disposal system may be universally applied. The furnace bottom is but one element of the design but it performs an important function.

Speaking generally, the slag-bottom furnace is advantageous to low capital costs, high heat release rates and direct firing of pulverized fuel. It is well adapted to burning the non-premium or cheaper grades of coal. The corrosion sometimes associated with overheating or deficient circulation in screen tubes is absent generally in the floor tubes of the slag-bottom furnace. The insulating nature of ordinary slag protects the water-cooled floor against overheating or faulty circulation, even with highly concentrated heat release rates and horizontal tubes in the bottom of the furnace. As this covering is maintained in place by gravity, the problem of floor cooling is comparatively simple. This permits great turbulence in the lower portion of the furnace and utilization of the entire volume very effectively. The slag bottom being practically hermetically sealed, eliminates the air leakage usually associated with ash hopper doors. Its attractiveness for direct firing is due to the stabilizing influence of the high temperature at the furnace floor which, aside from other advantages, prevents loss of ignition and the associated consequences. Under such conditions the danger of furnace explosions is greatly reduced. With direct or unit-mill firing the relation of primary air to coal is so limited, from a control standpoint, that the conditions are unfavorable to stable combustion, especially at light loads. The strong ignition conditions afforded by the high temperature of the slag bottom largely overcomes this deficiency and greatly increases the flexibility of control and range of rating of direct-fired installations.

A variety of arrangements of the slag-bottom furnace are in use, some of the continuous-flow type, others of the intermittent tapping type but with the former predominating in the more recent installations. The type of slagging arrangement affects the system of ash disposal

and the two should be considered together, along with other factors pertinent to the design. One elemental difference between the continuous slagging bottom and the side-tap furnace is that in the former case, the slag flows out almost continuously and is then stored in the quenched condition in a tank or hopper from which it may be sluiced or conveyed periodically to the ash bunker or point of delivery. In the intermittent slag-tap type the ash is stored in the molten condition within the furnace and is periodically tapped, quenched and delivered to the storage point, usually by sluicing. With this intermittent arrangement tapping, quenching and sluicing must be closely coordinated and usually this operation requires about one hour out of the twenty-four. Consequently, the rate of flow is high during this brief interval and substantial capacity is required in the ash-handling apparatus and sluice-pumping equipment. There are other means of disposing of this slag than by sluicing but this method predominates. The rate at which the slag flows from the side tap opening of the intermittent type is determined largely by the nature of the ash and the combustion rate. As these conditions are not completely under control, disposal facilities must be provided sufficient to take care of the maximum slag flow that is likely to be experienced. This consideration affects the investment in ash-handling facilities very materially and results in a low capacity factor on this apparatus.

Power Requirements vs. Labor

With the continuous slagging furnace the slag opening is usually in the bottom of the furnace rather than the side, the ash flowing from this opening into a receiving tank where it is quenched and stored or it may flow into a continuous sluice-way. The objection to the continuous sluice-way is the cost of power for sluicing the quenched slag. In some instances, receiving tanks beneath the furnace bottom have been provided sufficient to retain a day's output of quenched slag which may be periodically sluiced or conveyed to the point of disposal. This affords probably the cheapest and most convenient method of slag disposal, since it requires little or no attention. The rate of delivery of the quenched slag to the sluicing system is under control, hence the investment in sluicing or conveying equipment for handling the slag may be held to a low level.

It might be mentioned that the sluicing of ash or slag is economical in the use of operating labor, at the expense of higher power consumption and greater maintenance, as contrasted with the use of electrically-operated dump cars. In any event the disposition of labor must be carefully considered and if spare labor is already available whose presence is required by periodic inspection of other apparatus, the idle time may be utilized without additional labor costs. It is not always possible to foresee these conditions but they are factors which often dictate the choice in alternate schemes of development. From the standpoint of capital cost and capacity requirements, the power demand during the sluicing period for the intermittent tap furnace may run as high as 200 hp, while the conveyor drive, dump car power or sluicing power for the continuous-flow type need not exceed 20 hp and is often less.

Aside from these considerations, due thought must be given to the question of load range, fuel supply and factors which may impose operating limitations unless appropriate design is provided. With an adequate arrangement of either the side-tap or the continuous-flow type furnace, there is almost no limitation on the ability of experienced operators to maintain fluid slag, even with light loads and high fusion temperature ash. In one major plant difficulties were experienced with slag removal on both the continuous-flow type and slag-tap type in the early stages of operation, even at fairly high combustion rates. In recent years, however, these units

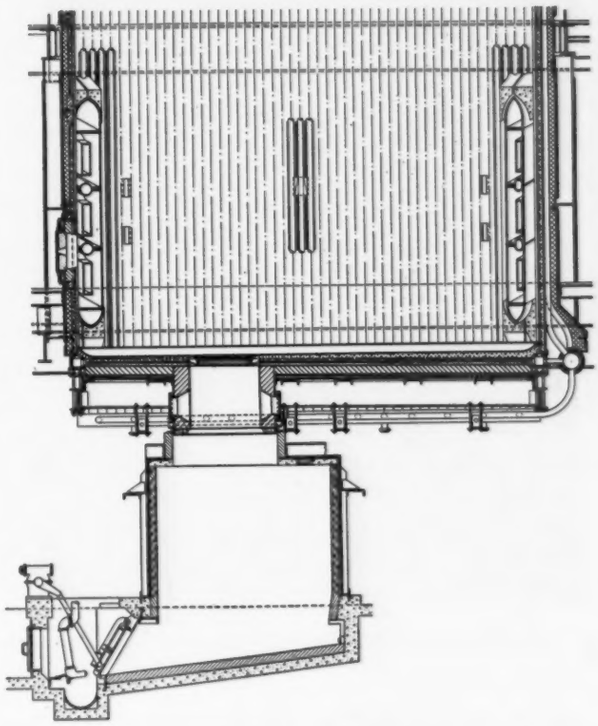


Fig. 2—Typical arrangement of furnace bottom for continuous slag disposal

have been operating in intermittent service with a shutdown of from 6 to 10 hr during the midnight watch, without any slag problem and with a wide variety of fuels and operating conditions. The proper adjustment of burner dampers, mill air supply, furnace draft and excess air are important considerations in slag removal at low ratings or with intermittent operation. The fundamental requirements are well understood but experience is essential in the development of successful operating technique.

Disposal System Must Be Able to Cope with Load Conditions

Under prolonged light load conditions in the continuous slagging furnace the slag flow diminishes and the floor thickness increases until temperature equilibrium is established, resulting in substantially more slag in the furnace than would exist at higher ratings. When the rating is increased to a sustained high level the rate of melting of this slag becomes more rapid and is supplemented by the melting of accumulations on the water walls, boiler tubes and other projections so that the flow rate increases rapidly until equilibrium again is restored. Unless the design of the receiving tank, sluice-way or conveyor system is adequate to meet this condition the disposal system becomes completely blocked with im-

properly quenched slag, rendering the unit unduly sensitive to range of rating.

While in the majority of installations of the continuous-flow bottom, the slag opening is in the floor of the furnace, this principle may also be utilized with the opening in the side wall. This arrangement has attractive advantages where head-room is limited, in which case the



Fig. 3—Looking toward bottom of tangentially-fired furnace under load

furnace floor may be at the basement elevation with a receiving pit or sluice-way for quenching below the slag opening. The quenched slag may be elevated by mechanical or hydraulic means for final disposal. Another method of handling quenched slag from the continuous-flow type where head-room is limited, is by quenching into the submerged receiving end of an inclined conveyor and delivery of the quenched slag into an elevated hopper adjacent to the boiler. The quenched slag may be conveyed by dump car or other appropriate means to the ash bunker.

In all cases it is preferable to have the slag-discharge opening sealed against the entry of cold air, as this usually interfers with the free flow of the material. The low conductivity and low specific heat of slag is an advantage, from an insulating standpoint, but at the same time imparts somewhat unfavorable characteristics, from the standpoint of disposal. The surface chills and solidifies rapidly, which interfers with the free flow and consequently necessitates shielding from cold air currents. At the same time, however, the application of heat restores fluidity so that by the induction of hot gas through the opening with the slag, fluidity may be readily restored or maintained. Experience has indicated that adequate cooling of the lip or crest of the slag opening is an important consideration, from the standpoint of maintenance, availability and ease of handling. In some installations of the continuous-flow slag bottom the slag opening is made large enough to afford a convenient means of entry into the furnace.

Enclosing the slag disposal system is also in the interest of safety, since it protects the attendants from slight explosions which occasionally accompany the quenching of slags high in iron sulfide. The water-cooled bottom permits somewhat more rapid cooling of the slag floor than the uncooled bottom when a boiler is taken out of service and probably affords more uniform temperature distribution during the cooling period. Rapid cooling may be tolerated in low-pressure boilers but it is a condition to be avoided in high-pressure boilers, as it may result in joint leakage. However, the rate of cooling is under the control of the operator and is of little significance in the choice of a furnace bottom and ash-disposal system.

Cooled Bottoms Have Overcome Earlier Expansion Troubles

In the early experience of uncooled slag-bottom furnaces of the intermittent-tap type, some trouble resulted from pushing out of the side walls due to the swelling tendencies of the slag bed with successive cycles of heating and cooling. In some instances this was attributed to segregation of the heavier constituents of the ash, such as sulfide of iron, and in general was quite troublesome. It was necessary, occasionally, to dig away the slag adjacent to the walls during boiler outage and provide a channel around the edge of the slag bed to accommodate the expansion. These problems have been generally overcome and are absent in the modern designs of cooled bottoms. It is likely that the characteristics of the early mills were contributing factors, especially in the coarseness of the particles of iron sulfide introduced into the furnace. In some of the modern mills a large portion of sulfide of iron is rejected through the discard spout, which is advantageous to both the mill and furnace. With some American coals as much as 50 per cent of the sulphur in the coal has been accounted for in this discharge. In

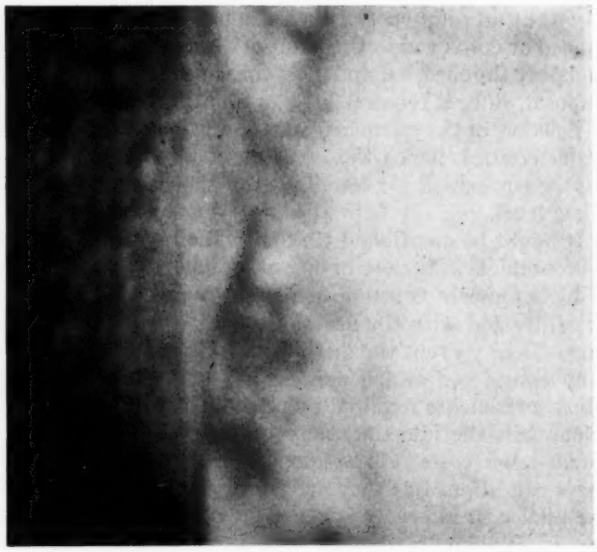


Fig. 4—View taken across same furnace as in Fig. 3 at side and level of burners. The bright spot in the center is a multiple reflection from the camera lens

one plant, from each 12-ton mill, the discarded pyrites at times amount to as much as 300 lb per hr.

In the bottom covering of water-cooled furnaces various practices prevail as to the materials used. In some in-

stances, chrome ore is spread over the tubes, followed by a layer of stoker ash. Gradually this stoker ash is fused and flows out in the course of operation, being displaced due to selective action by the higher fusion constituents of the fly ash. In other installations stoker ash is the only covering used and in all cases it is essential to maintain a sufficient slag depth to protect the upper sides of the floor tubes from the excessive temperature due to the intense combustion at the bottom of the furnace. The depth of the slag bed may be readily maintained by the height of the water-cooled slag lip above the elevation of the furnace floor. In some of the air-cooled furnace bottoms more elaborate covering is necessary, consisting of chrome ore in graduated sizes, together with various sealing materials, to prevent the slag from penetrating the air-passages. These bottoms have given very successful service and credit is due for the ingenuity in their design. The principal disadvantage of this type is the need for high pressure ventilating air and its chilling effect upon the slag bed, above which it is discharged into the furnace.

Large Percentage of Fine Ash Caught in Slagging Bottom

The slag-bottom furnace catches a large part of the ash in the fuel, some recovering more than 50 per cent, which accounts for a lower stack emission than would otherwise result. Because of selective melting, the lower fusion ash flows from the furnace, while that which passes through the gas passages has less slagging tendency due to the higher fusion temperature. In various localities the slag has potential value and aside from commercial uses it makes an excellent fill, due to its high density and cleanliness.

In some plants the fly ash from dust collectors is delivered to the furnace where it is fused and disposed of along with the slag. The hardness of most furnace slags



Fig. 5—Water sprays for quenching slag as employed on several large units. These sprays are located beneath the slag opening

exceeds that of glass and while attractive as abrasives, this aggravates the maintenance problem on the ash-handling equipment. Water cooling affords a thinner floor than the conventional air cooling but the latter might be equally attractive and cheaper if the arrangement could be accommodated in the design without the introduction of excessive duct work.

Troubles with the early installations may be regarded as the price of progress and this experience paved the way for advances in furnace design which far overshadow the incidental expenses.

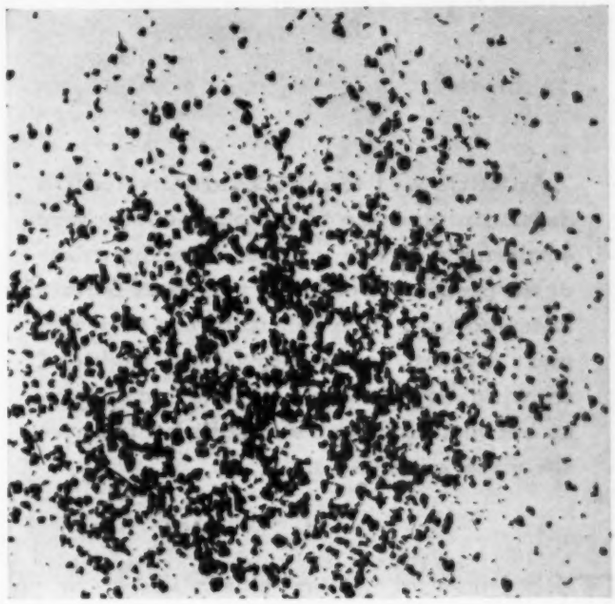


Fig. 6—The slag after being quenched by the sprays is about the size of buckshot

The accompanying illustrations are typical for a steam generating unit having a continuous slagging furnace and the photograph on the cover of this issue represents a view of the molten slag as it drips through the floor opening just before being quenched.

A.S.M.E. Nominations

Nominations for officers of the American Society of Mechanical Engineers for 1939 were announced by the Nominating Committee during the recent Semi-Annual Meeting of the Society at St. Louis. The nominees which are to be voted on by letter ballot of the entire membership, on or before September 27, are:

President, A. G. Christie, Professor of Mechanical Engineering, Johns Hopkins University, Baltimore, Md.

Vice-Presidents, Henry H. Snelling, Snelling & Hendricks, Washington, D. C.

W. Lyle Dudley, Vice-President, Western Blower Company, Seattle, Wash.

James W. Parker, Vice-President and Chief Engineer, Detroit Edison Company, Detroit, Mich.

Alfred Iddles, Application Engineer, Babcock & Wilcox Company, New York, N. Y.

Managers, Clarke Freeman, Vice-President, Mfrs. Mutual Fire Ins. Company, Providence, R. I.

Willis R. Woolrich, Dean of Engineering, University of Texas, Austin, Texas

William H. Winterrowd, Vice-Chairman, Franklin Railway Supply Company, Chicago, Ill.

The Selection of Economizers and Air Preheaters

By J. H. SENGSTAKEN
The Superheater Company

An outline of the procedure involved in determining when to install heat recovery equipment in the form of an economizer or air preheater, or both. Various factors entering into such selection are reviewed and typical calculations are included showing how to evaluate the net return as between different sizes offered to meet the given conditions.

THE process of selecting economizers or air preheaters, or both, for boiler application involves careful study and evaluation if the correct size and design are to be purchased. The factors involved are the same for both types of equipment, whether to be applied to either new or existing boilers. In the latter case, local conditions frequently outweigh all other considerations, limiting the selection to one or the other type and thereby greatly reducing the amount of work in the process of evaluation.

One example of such a case is that in which heat recovery equipment is to be applied to a stoker-fired boiler. The accompanying curve, Fig. 1, reproduced from a paper by Messrs. Dillon and Engle on "The Economics of Preheated Air for Stokers,"[1] shows the average cost of maintenance with preheated air of varying temperatures from data supplied by thirty-three operating companies on one hundred and twenty-two underfeed stokers, and indicates that above 300 to 350 F air temperature these costs increase rapidly. Considering that the air temperatures stated are maximum, those at normal loads will be lower and the increase in efficiency represented by the heat extracted from the flue gases will be correspondingly less.

It is generally true that if alterations and additions to an existing stoker-fired installation are at all justifiable the optimum overall efficiency with an economizer is higher than that which is practical with an air preheater. The installed cost of an economizer that does not require changes to forced-draft fans or additional air ducts is not greatly different from that of the air preheater under this condition. The desired efficiency generally can be obtained with the economizer and this is usually done without giving serious study to the application of air preheaters. With other methods of firing and with other fuels, however, both types of heat recovery equipment should be considered.

For the past decade increased fuel costs and higher rates of evaporation have been responsible for improvements in design that have rendered the attainable efficiency of new steam generating units higher than previously. Hence, economizers and air preheaters are considered in all but a small percentage of new installa-

[1] Published in *Transactions* of the American Society of Mechanical Engineers, December 1934.

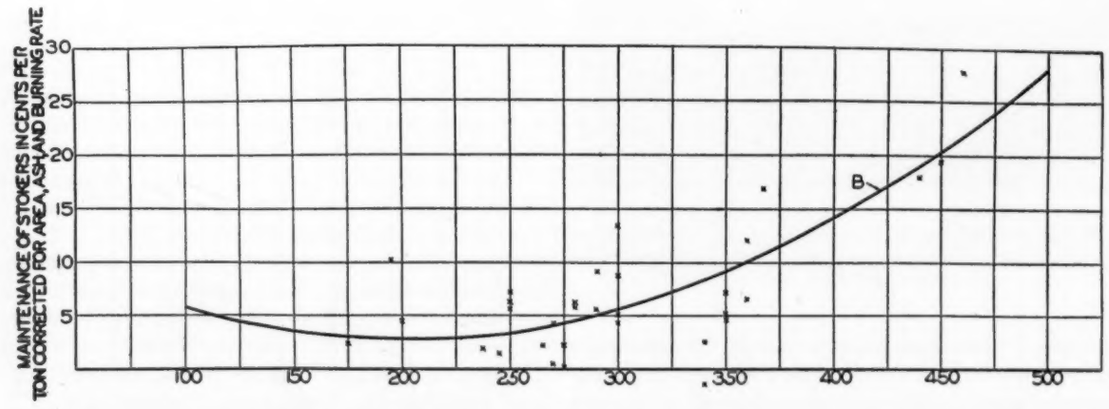


Fig. 1—Maintenance costs with stokers operating at various air preheat temperatures, based on 122 stoker installations

tions either as alternates or arranged in series. Higher initial steam pressures and higher steam temperatures, together with regenerative feedwater heating, have resulted in improved station performance and increased efficiency of the prime mover. But both the higher pressures and the higher temperatures tend toward increased exit gas temperatures which render the installation of heat recovery equipment more important. With increased bleeding of steam from turbines the boiler feedwater temperature has gone up and the possible heat recovery in economizers has been correspondingly reduced. The air preheater has therefore taken on added importance, and where fuel and firing conditions permit, it is designed to recover a much greater proportion of the waste heat than heretofore. Thus, where previously, with gas, oil or pulverized coal firing, a maximum of 400 to 450 F air temperature was considered good engineering, a number of recent installations have been made with the maximum air temperatures between 550 and 600 F.

When a study of the application of economizers and air preheaters is made for existing boilers the load curve is usually pretty well known. In the case of new units the predicted load curve should be prepared and the number of hours per year at each load determined. From this the number of steam generating units to be installed is determined. As the knowledge of boiler design, water circulation, control of furnaces, etc., has increased the availability of steam generating units has become extremely high. Records indicate that a number of such large high-pressure, high-capacity units have never had an emergency shutdown. Thus, the designers of power stations are enabled not only to reduce the number of spare units required but may also avail themselves of the lower installed cost per pound of steam by employing a minimum number of large high-capacity boilers instead of a greater number of smaller units. The gross annual saving due to waste heat recovery equipment is in direct proportion to the number of hours per year it is in operation. This is dependent not only on the load factor but also on its availability or use factor, which obviously should be equal to that of the boiler itself.

Exit Gas Temperatures Important

While the normal or average evaporation in either new or old boilers together with the unit cost of the fuel will determine to a large extent the optimum efficiency, the exit gas temperature at the minimum loads should also be calculated. It is sometimes discovered that the exit gas temperature under the latter conditions is so low that corrosion of an economizer or air preheater will take place. To avoid this, the exit gas temperature at the lowest load should be increased by either reducing the size of the heat recovery equipment and thus reducing the efficiency over the whole load range, or in the case of the preheater, by recirculating the air at low loads. In the latter case the efficiency can be maintained at the desirable point at normal loads.

It has frequently been shown that draft loss through economizers and air preheaters pays for itself in increased heat transfer and, therefore, for a given investment in higher net returns. For this reason the draft loss of the complete steam generating unit may be made as high as is practical at the average evaporation without exceeding reasonable limits at peak load. It should be borne in mind that each design or type of economizer or air preheater has a different character of resistance to gas and air flow, and that a high resistance is not necessarily conducive to the best net efficiency. Each kilowatt-hour of electric power consumed in fan power must be paid for and charged against the saving produced in annual fuel costs.

The initial cost of any heat recovery apparatus includes, in addition to the price of the economizer or air preheater itself, the extra cost of any material that would not be required if the boiler alone were installed. Thus. additional flue work, air ducts, supporting steel, increased induced- and forced-draft costs and additional building costs, if any, should be added to the cost of the equip-

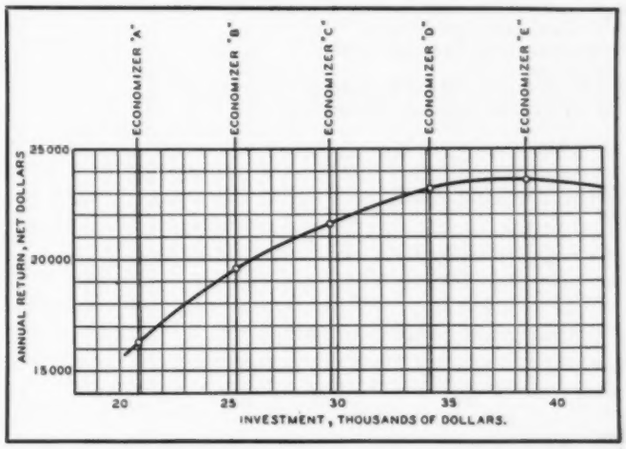


Fig. 2—Curve illustrating sample selection of an economizer

ment itself. These costs will vary with each type and design considered.

The gross savings produced by the heat recovery equipment are represented by the difference in the fuel cost when operation is considered at the efficiency of the boiler alone and at that of the complete unit including heat recovery equipment. The net savings are those remaining after the fan power cost, maintenance and fixed charges have been subtracted from the gross savings. The fan power chargeable against the heat recovery equipment is the difference between that required for the complete unit at its exit gas temperature and that for the boiler at the temperature leaving it. In the event that a stack alone could supply sufficient draft for the boiler alone the additional stack cost over that required for induced-draft operation should properly be added to the boiler cost.

Maintenance and fixed charges are ordinarily taken at 12 to 18 per cent depending on the design of equipment being considered and the previous experience with it.

When only one type of equipment is being considered, a family of curves should be drawn for the design submitted by each manufacturer, the curves showing the net annual return in dollars being plotted against total additional investment. Each curve should apply to a group of units, each with the same cross-section but of varying heating surfaces. In this manner curves showing the net return of a group of units with low gas flows and low draft losses may be compared with similar groups at successively higher rates of flow. Since the

cost of the additional draft loss due to the higher gas flows is included in calculating the curves, the desirable heat recovery may be easily selected.

The same procedure may be followed when both economizers and air preheaters are to be used and the most satisfactory combination selected.

As an illustration, a curve is given in Fig. 2, showing a sample selection for an economizer. For the sake of simplicity only one of the family of curves has been plotted for an economizer to be applied to an existing stoker-fired boiler. The following conditions have been assumed:

Average load—150,000 lb of steam per hour for 6000 hr per yr
Maximum load—200,000 lb of steam per hour
Steam conditions—450 lb pressure, 750 F final steam temperature, 212 F feedwater temperature
Fuel—13,500 Btu coal costing $5 per ton of 2000 lb
Boiler exit gas temperature, F—700
Boiler draft loss, in. of water—1.50
CO_2, per cent—14
Boiler efficiency without economizer, per cent—73.5
Fan power cost per kwhr, cents—0.50

Five sizes of economizers of a standard commercial design are compared, each with the same number of tubes in width, and all of equal tube length, but of successively greater number of rows in height. As the gas areas are the same in all cases, the rates of gas flow and therefore the rates of heat transfer are comparable. The draft losses are in direct proportion to the number of rows high. Table I gives the performance of the economizers compared with the boiler alone.

TABLE I—COMPARATIVE PERFORMANCES FOR FIVE DIFFERENT ECONOMIZERS AND BOILER ALONE

	Bare Boiler	A	B	C	D	E
Economizer heating surface, sq ft	...	4620	6150	7700	9240	10,780
Exit gas temperature, F	700	480	435	400	370	355
Overall efficiency, per cent	73.5	79.30	80.50	81.40	82.20	82.60
Draft loss, economizer, in. water	...	1.45	1.84	2.22	2.60	2.95
Draft loss, boiler, in. water	1.50	1.50	1.50	1.50	1.50	1.50
Draft loss, flues, etc., in. water	0.50	0.50	0.50	0.50	0.50	0.50
Total draft at I.D. fan, in. water	2.0	3.45	3.84	4.22	4.60	4.95
Fan power required, kw	58.0	75.2	78.4	81.8	85.4	89.8
Fan power chargeable to economizer, kw	...	17.2	20.4	23.8	27.4	31.8

It has been assumed that an existing induced-draft fan will be removed and a new one purchased capable of handling the increased draft. The fan efficiency has been taken at 60 per cent and the motor and line efficiency at 85 per cent in all cases, and the additional power over that required for the existing installation is charged against the economizer.

In Table II the net dollars return for each economizer is calculated and in each case the annual fuel cost is calculated from the overall efficiencies shown in Table I.

TABLE II—ESTIMATED COSTS

Economizer	A	B	C	D	E
1. Estimated cost of economizer installed	$13,860	$18,170	$22,300	$26,500	$30,730
2. Estimated cost of fans and motors installed	4,000	4,200	4,400	4,600	4,800
3. Estimated cost of flues and steelwork	3,000	3,000	3,000	3,000	3,000
4. Total cost of installation	$20,860	$25,370	$29,700	$34,100	$38,530
5. Cost of fan power chargeable to economizers from Table I	$ 516	$ 606	$ 714	$ 822	$ 954
6. Carrying charges on installed cost of installation at 15%	3,130	3,795	4,460	5,120	5,780
7. Total annual charges	$ 3,646	$ 4,401	$ 5,174	$ 5,942	$ 6,734

The costs given for the various economizers are approximately correct, and the installed fan costs are made from estimates of prices for the actual conditions set up. The costs of flues and steel work are estimates but since they are constant in all cases their accuracy does not affect the result.

The carrying charges of 15 per cent include amortization, regular maintenance and repair.

Table III below shows the net annual savings from each of the economizers.

TABLE III—NET ANNUAL SAVINGS

Economizer	A	B	C	D	E
1. Cost of fuel per year for boiler alone	$274,000	$274,000	$274,000	$274,000	$274,000
2. Cost of fuel per year with economizer	254,000	250,000	247,300	244,800	243,600
3. Gross dollars annual saving	$20,000	$24,000	$26,700	$29,200	$30,400
4. Less item 7, Table II	3,646	4,401	5,174	5,942	6,734
5. Net annual dollars saving	16,354	19,599	21,526	23,258	23,666
6. Additional investment required for each larger size economizer		4,440	4,400	4,400	4,430
7. Additional net return each larger size economizer		3,239	2,033	1,632	408
8. Per cent net return on additional investment		73.0	46.2	37.1	9.2
9. Per cent net return of economizer on total investment	78.4	77.4	72.8	68.2	61.4

Item 9 of Table III indicates that any one of the economizers will pay an attractive return on the investment, but item 8 indicates which one should be selected. The net returns on the incremental additional cost for each succeeding larger size show an ample percentage up to economizer "D." In economizer "E" the net return on the additional investment is only 9.2 per cent which is not sufficiently high to warrant the expenditure. If the table were carried out further it could be shown that while the return on the total investment for larger sizes might still be attractive the incremental net return would become a negative quantity. In other words, the additional carrying charges would become greater than the additional fuel saving.

As stated previously, the example given was selected for its simplicity. In considering an air preheater alone the only differences would be the addition of the net cost of the forced-draft fan power or the difference between that required with and without the preheater, and in addition to the flue cost, that for the extra air ducts required. The type of calculation illustrated is applicable equally well to combinations of economizers and preheaters and similar curves can be drawn to obtain the best efficiency.

While the conditions set up for the calculation of the tables may be thought ideal they are by no means uncommon. The advantage of making a study of this kind for any installation is that it will not only indicate how much heat recovery should be purchased but whether or not local conditions will, from the financial considerations, permit its installation at all. One example of the borderline plant is, of course, the steam standby station for a hydroelectric plant. Studies for such cases have indicated that more often than not heat recovery has been found unwarranted from the standpoint of the return on the investment but in some pulverized coal installations air preheaters have been installed to reduce the cost of pulverizing and to increase the combustion efficiency.

Stability of Colloidal Fuels

By C. H. S. TUPHOLME

This is a review of the results obtained from investigations by the British Fuel Research Board on the suspension of pulverized coal in oil. The curves showing the rate of settlement of the coal were obtained in the laboratory by the pendulum method and reveal anomalies in the viscosities of some of the oils regarded as suitable for this purpose.

PART of the British program for national self-sufficiency is the investigation of every possible means for making the utmost use of native coal deposits, such as hydrogenation of coal and low temperature tars to obtain motor spirit. Another branch of this work is the growing use of suspensions of finely divided coal in oil—a fuel which has been used successfully in several ocean-going liners as well as on land. Investigations for improving the stability of such colloidal fuels are now being undertaken by the Fuel Research Board, which is a division of the Department of Scientific and Industrial Research.

The laboratory methods available for investigating the mechanism of stability in suspensions of pulverized coal in oil have been greatly reinforced by the evolution of the pendulum method for assessing the rate and extent of settlement and by a simple method that can be used for revealing anomalies in the viscosities of some of the oils suitable for making suspensions of the requisite stability.

The pendulum enables a very accurate comparison of the stabilities of different suspensions to be determined over a wide range. Fig. 1 gives an idea of the reproducibility of results. It shows the amount of sinkage after different intervals of time shown in four different samples prepared by mixing the same coal and oil in the same proportions under conditions that differed only slightly if at all, from one sample to the next. The observations were

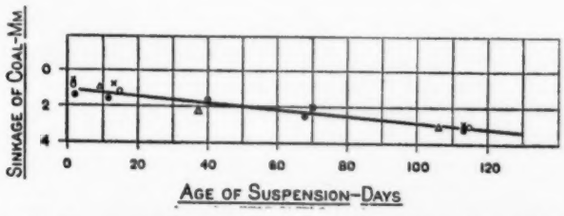


Fig. 1—Curve illustrates reproducibility of results with four different samples of the same mixture

carried out at 25 C., and, taken generally, conditions were not as favorable to the reproduction of results as they might have been.

Different types of curves that can be obtained are shown in Fig. 2. A change in the period of the pendulum caused by the sinkage of suspended material would not be expected to appear if oil alone were examined, and one of the curves shows this for two quite different oils. The three other curves show the wide range over which the rate of settlement may vary from one suspension to another. The most important parts of these curves are the earliest portions, from which it is possible to obtain an estimate of the average initial rate of settlement of the suspended coal. This rate is the average downward velocity of all the particles of coal in the mixture contained in an infinitely deep vessel, or, in the practical examples in a vessel of finite depth before the accumulation of coal on the bottom has become great enough to exert an appreciable effect. The determination of the average

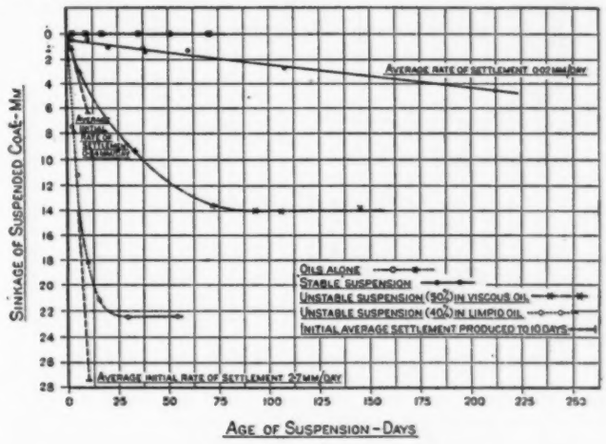


Fig. 2—Illustrating stable and unstable suspensions

initial rate of settlement is the best of the several different ways that the method makes available for comparing the stabilities of suspensions. Three of the curves of Fig. 2 have average initial rates of settlement of 0.02, 0.64 and 2.7 mm. per day; the four samples of Fig. 1 have a slightly greater stability than the best of the three mixtures of Fig. 2 with an average initial rate of 0.017 mm. per day.

As the examination of the samples is carried on over a long enough period, the average rate of settlement decreases progressively and becomes zero when settlement is complete. Two of the curves in Fig. 2 show this. But this slackening of the rate of settlement and the point at which it becomes zero depend not only on the properties of the suspension but also on the depth of it in the vessel in which it is contained, so that if the life of the suspension is to be used as a comparative measure of stability, the life must be reckoned on a pre-selected depth of the suspension. The lives—that is the lengths of time before settlement is complete—of the two samples that are shown in Fig. 2 to have been examined to completion are 85 and 27 days.

These two samples indicate that the total amount of settlement has occurred to different depths, although for each the original depths of the suspension was the same. This is because one contains 40 per cent of coal and the

other over 50 per cent and when settlement is complete these different amounts of coal, distributed initially over the same volume, will have moved through different distances in order finally to be packed in the same way.

Oils that have proved suitable for colloidal fuel have been shown in a general way to display abnormalities in viscosity in that if the viscosity is determined at progressively decreasing rates of shear, it shows a progressive increase in magnitude. This has been ascribed to the existence in the oil of some kind of structure, and may be due to incipient gelation. Many liquids with similar properties are known, and those that display this property as a reversible sol-gel transformation, with the pas-

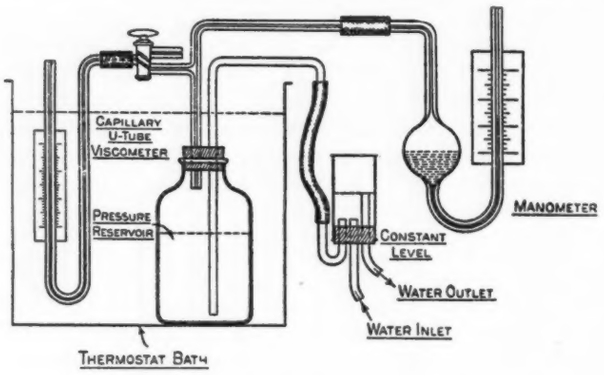


Fig. 3—Outline of test apparatus

sage from gel to sol attainable merely by agitation, are designated "thixotropic." In such liquids the effects due to structure lessen or disappear on agitation, and also on heating.

The apparatus used is shown in Fig. 3. A constant-level device is connected to an air-reservoir in such a way that the air can be stored under a small and constant pressure. The oil, after vigorous agitation, is introduced into a capillary U-tube; alternatively, and more conveniently, it is introduced into the U-tube first and warmed afterwards in a bath at a temperature high enough to resolve any structure that may have been present; 50 C. seems to be a high enough temperature as far as the investigation has gone. The U-tube is then put into a constant-temperature bath at 25 C., and communication is established with the pressure-reservoir, which is also kept in the constant-temperature bath; from the moment when the U-tube and pressure reservoir are placed in communication, the time is noted for a small displacement of the oil in the U-tube. Contact with the pressure reservoir is then interrupted, the oil is allowed to level off with both limbs of the U-tube open to the atmosphere, and then, after the lapse of a suitable period, a record of the time for the displacement of the oil in the U-tube is again taken.

The converse of the increase in viscosity with age, that is, the decrease in viscosity by mechanical agitation—the resolution of structure on which the definition of thixotropy is based—was investigated with the more suitable of these two oils. The initial time of displacement of 33 sec at 19 C. had become 41 sec after an hour and a quarter. When this observation had been made the oil was immediately restored to a level position and redisplaced; this procedure was repeated several times. The agitation to which the oil was subjected was merely the reciprocal displacement of it in the U-tube during the course of making observations. The final time of displacement—about 32 sec—is roughly in agreement with the time of 33 sec shown by the oil at the very commencement of the investigations on it. A similar decrease in viscosity could be brought about when, after six hours at 19 C., the time of displacement had increased to eight minutes.

If the curves showing "stable" suspensions in Figs. 1 and 2 are examined, it will be seen that whereas these curves follow a very gentle and practically constant slope for the greater part of their courses, they show during the first few days a much greater rate of settlement. This is attributed to the same phenomenon, the dependence of viscosity on the immediate previous history of the oil. When the specimens are mixed and introduced into the tubes in which they are to be examined, they are subjected to very thorough and vigorous agitation. The structure contributing to stability in the suspension is thus destroyed, and settlement takes place correspondingly rapidly; but within from two to four days, the structure has redeveloped, the rate of settlement is greatly reduced; and the curve displays the described change in direction. This can be easily understood if the pendulum method of tracing the course of settlement is regarded as a modification of Stokes' method of viscosity determination; the falling sphere is replaced by minute particles of coal, and by virtue of the very low rates of shear that they set up in falling, their motion reflects the changes in the viscosity of the oil.

The diagrams are reproduced by courtesy of H. M. Director of Fuel Research.



Design of High-Capacity Boilers*

By JOHN VAN BRUNT, Vice President
Combustion Engineering Company, Inc.

The author briefly reviews the development in designs leading up to the modern high-pressure high-capacity steam-generating unit and shows the limitations of the earlier types that render them unsuited to high-capacity requirements. Problems entering into the design of such units are discussed and curves are incorporated covering certain basic relations. Typical units are shown and the performances of several are included.

POWER boilers may be divided into a number of groups, depending upon the basis of the classification. In whatever class a boiler may be arbitrarily placed the designer must conform to the A.S.M.E. Boiler Construction Code. Within the limits and rules of the Code the designer has much latitude. In fact, it is highly probable that a boiler could be (and possibly has been) designed that would conform to the Code requirements but which, none the less, would be unsafe.

There are many boilers in use today that are satisfactory and safe at the capacities at which they are operating but which, if operated at twice the present capacity, would be unsatisfactory and possibly unsafe.

This paper will not attempt to more than outline the general procedure followed in arriving at the final design of large or high capacity steam generating units as exemplified in some of the recent installations of this class. The problems of design for 1400-lb pressure boilers are much the same as those involved in boilers for lower pressures. The details differ only in degree.

The modern high capacity steam generating unit is more than a boiler, it is a completely coordinated and balanced assembly of furnace, superheater, boiler, setting, economizer, air preheater, forced- and induced-draft fans and fuel-burning equipment. The last may be a stoker, oil burners or pulverized coal equipment comprising mills with means for drying coal, feeders, burners, coal piping, etc. Also, there are automatic regulators, boiler-feed pumps and all auxiliaries necessary for the control and operation of such a unit.

The two outstanding developments without which the present high-capacity units would not be possible are pulverized coal and the water-wall furnace. Stokers having a fuel burning capacity of 25 to 50 tons an hour are hardly practicable even though mechanically possible. Multiple-retort stokers have been built to burn 25 tons per hour, but operation at that rate is not entirely satisfactory. Chain-grate stokers have been built up to 575 sq ft of grate area, but it is doubtful if this area can be increased by more than a few per cent.

Refractory furnaces cannot be used with pulverized coal except at low combustion rates and therefore are not considered for large units.

In large high-pressure units the boiler provides only a small part of the total heat-absorbing surface, from 6 to perhaps 25 per cent. It, however, plays an important part in the final design and forms a framework around, and within, which the unit is designed.

Figs. 1 to 4 illustrate well-known types of boilers which have given satisfactory service for many years, but for high capacity units these types are inadequate because of faulty circulation.

The straight-tube boiler shown in Fig. 1 has given good service up to evaporation rates of 6 or 7 lb of steam per square foot of heating surface. At these rates the upper tubes are likely to be partly dry, water flowing along the bottom of the tubes and steam in the upper half. Sometimes this does no harm; in fact many boilers have operated many years under such conditions.

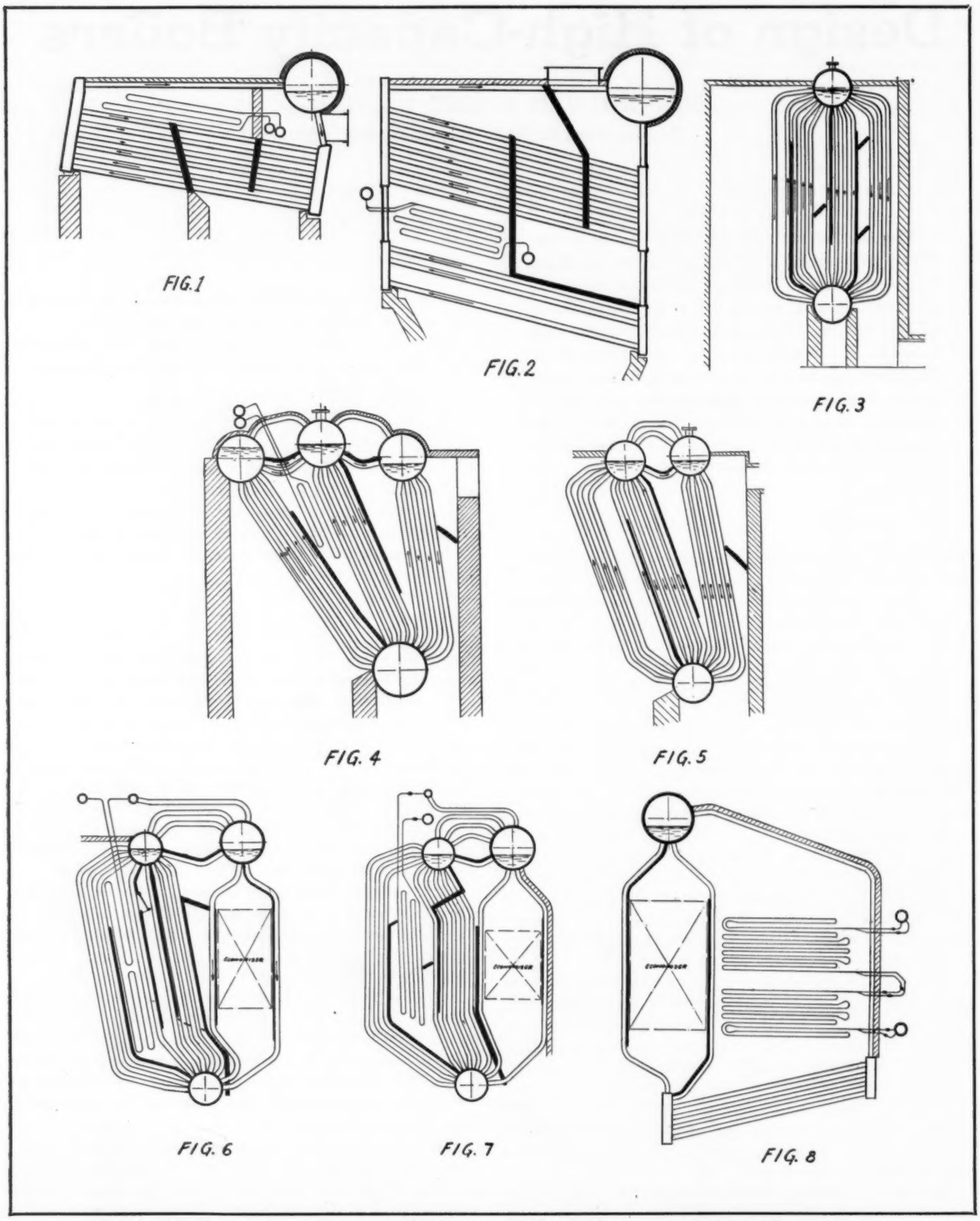
Fig. 2 shows a boiler of the same type with interdeck superheater. This boiler is subject to the same fault in circulation, and when operated at steady high loads, half or more of the tubes in the upper bank may be partly dry. Rope scale may form along the water line. Corrosion may also occur along the water line, and in high-pressure boilers at the top surface of the dry tubes.

Figs. 3, 4 and 5 show three well-known bent-tube types. That shown in Fig. 3, while good from the standpoint of circulation, may be a wet steamer when operated at high capacity because of turbulence in the drum. Steam quality is largely dependent on boiler water quality.

Fig. 4 represents a four-drum design which has had wide use and has given satisfactory service in numerous plants for many years. However, the circulation in this boiler leaves much to be desired. Insufficient provision is made to unload the upper front drum. Turbulence in the center or steam drum limits the capacity obtainable with good steam quality.

Fig. 5 shows a three-drum boiler, also a well-known type. This has a well balanced circulation, which in general follows the arrows. The front drum permits separation of steam and water, the steam passing to the rear drum through the upper circulators and the water passing down through the second tube bank. The rear drum is not turbulent, as little steam is generated in the third or rear bank of tubes. Steam is taken from the rear drum. Figs. 6 and 7 are modifications of this type in which the rear bank of tubes is replaced by an economizer located between two rows of tubes connecting the steam drum and the lower drum.

* A paper delivered at the Semi-Annual Meeting of the A.S.M.E. at St. Louis, June 21, 1938.



Figs. 1 to 8—Comparing earlier types of boilers, unsuited to high-capacity service, with recent types

Fig. 8 represents a modification of the straight-tube type which has proven satisfactory for high-capacity operation. Only three, or at most four, rows of inclined tubes are used; if more are employed there is danger that the circulation in the upper rows would be deficient, as in the boilers shown in Figs. 1 and 2.

The addition of water walls to any of the boilers just described does not materially affect the circulation but with the single-drum boilers the added turbulence, due to the discharge of the water from the water walls, makes the problem of obtaining dry steam somewhat more difficult.

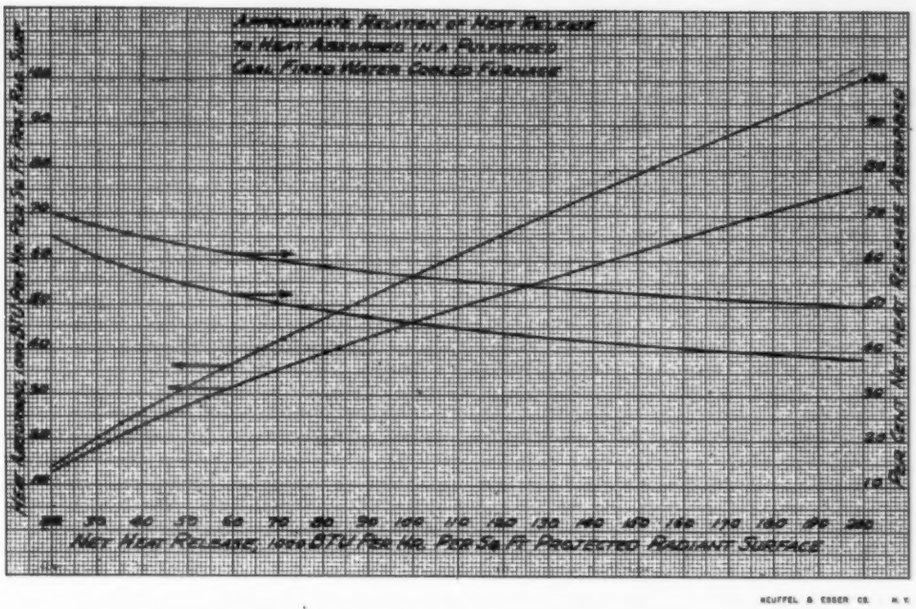


Fig. 9—Relation of heat release to heat absorption in water-cooled furnace

In boiler plants of twenty to thirty years ago it was the usual practice to install spare capacity to the extent of 10 per cent or more. In some cases the formula was 10 per cent plus one boiler. Today, one boiler will produce as much, or more than ten such boilers and frequently no spare capacity is provided. Naturally, today's boilers must be exceptionally dependable and must be available the maximum possible number of hours annually. It is felt that such boilers should and must be designed with certain objectives or requirements, among which are the following:

Desired output for as long a period as required.
Maximum economy—not necessarily efficiency.
Required steam temperature over a specified range of capacity.
Control of superheat.
Dependability or maximum availability.
Freedom from circulation troubles.
Clean steam.
Low heat loss from radiation.

An inquiry for a boiler of this type usually specifies, among other requirements:

Capacity—continuous and peak.
Pressure at the turbine throttle or superheater outlet.
Steam temperature at normal load and at peak load, or at various points of the load curve.
Feedwater temperature.
Kind of coal, with analysis, moisture, ash fusion temperature, heating value and grindability.
If any unusual load requirements are desired, such should be specified.

The inquiry may state that the unit shall be fired with pulverized coal and a slagging bottom furnace may be also specified.

This constitutes the data from which the boiler is designed.

Recognizing that slag on boiler or superheater surface is one of the principal causes of boiler outage, design should be based primarily on a furnace with water walls of such volume and area that the temperature of gas leaving the furnace or entering the boiler tubes is below the fusion temperature of the coal ash. There may be a considerable spread between the fluid temperature and the softening temperature and where possible the design should be arranged for the lower temperature.

This is not difficult when the ash-softening temperature is above 2200 or 2300 F, or when the required steam temperature is not over 750 to 800 F. Where the ash-softening temperature is 1950 F to 2000 F and the required steam temperature is 900 F or above, the problem is difficult, if, indeed, it can be completely solved.

To obtain 925 F steam temperature the gas entering the superheater must be close to 1900 or 1950 F. If the superheater is shaded in part from furnace radiation by three or more rows of boiler tubes, the gas entering the boiler tubes must be 2000 to 2050 F. In such cases ready access must be had to both the front rows of boiler tubes and the superheater for soot blowing or hand lancing for the removal of slag or crusted ash.

Having determined what temperature of gas is necessary for the desired superheat and the temperature of gas desired leaving the furnace, the latter is then proportioned to provide sufficient water-cooled surface to reduce the temperature of gas leaving the furnace to the required figure.

The first furnace dimension determined is the width. Inasmuch as the major factor in boiler cost is the drums, the width of furnace should be as small as possible consistent with the draft losses through the unit and also the permissible rates of steam release per foot of furnace or boiler width. If the unit is going in an existing boiler room, the spacing of columns and height of roof trusses may limit the width or height of the proposed unit so that a compromise becomes necessary in the design.

In determining the amount of heat absorbed in the furnace, the relation of heat released per square foot of furnace wall surface to heat absorbed by such surface is needed; such relation is shown in Fig. 9. Obviously, this relation depends on a number of factors, among which are (*a*) effectiveness of surface, (*b*) cleanliness of surface, (*c*) distribution of flame in the furnace, (*d*)

velocity of gas in contact with walls, (*e*) luminosity, (*f*) area of flame surface, (*g*) temperature of flame and (*h*) excess air in furnace.

As an illustration of the effect of the method of firing, data are available to show that of two furnaces having the same volume, the same surface and the same heat release, one of them tangentially fired above a slagging bottom, the other horizontally fired or vertically fired above a slagging bottom, the gas temperature leaving the first named will be about 200 F lower than in the other one. A difference of 5000 to 7500 Btu absorbed per square foot of wall will account for this difference.

In selecting the proper curve of $\frac{\text{Heat Absorbed}}{\text{Heat Released}}$ consideration must be given to the various factors mentioned above, and in this selection experience and judgment play an important part.

The furnace dimensions having been determined and also the gas temperature entering the boiler, the size of superheater must next be ascertained. This involves the usual calculations based on mean temperature difference of gas and steam, mass flow of gas and transfer rates determined from experience, all of this with due regard to draft loss and accessibility for cleaning superheaters, the soot blower location, etc. If boiler surface follows the superheater, the temperature of gas leaving the boiler or entering the economizer is calculated.

In a pulverized coal installation preheated air is necessary to dry the coal in the pulverizing process; therefore, the economizer will be designed so that the gas temperature leaving it will be high enough to permit preheating the air to a temperature of 450 to 500 F.

Depending on the temperature of water entering the economizer, that leaving may be 50 to 100 F below the boiler saturation temperature. At high pressures this practice is safe. There is little probability of the economizer steaming unless an unduly long time is taken to raise the pressure. In such cases the feed valve should be opened and water allowed to flow into the boiler, and the boiler blown down.

Having determined the surfaces required in the furnace, boiler, superheater, economizer and air preheater, the next step is to assemble these units. The furnace and boiler must be coordinated so that the necessary downcomers and risers for the water walls can be installed with proper provision for expansion. Space within the boiler must be provided for the superheater and if steam temperature requirements necessitate, a bypass damper must be designed for superheat control.

If the economizer is placed within the confines of the boiler, the whole unit must be enclosed in a suitable setting or casing which must be as near gas tight as is possible.

Returning to the water-cooled or water-wall furnace, a few comments on design may be of interest.

The three factors necessary for complete combustion are time, temperature and turbulence—time for the combustion of solid particles of carbon, time for the oxygen in the furnace to make contact with the combustible matter; temperature sufficient to insure prompt

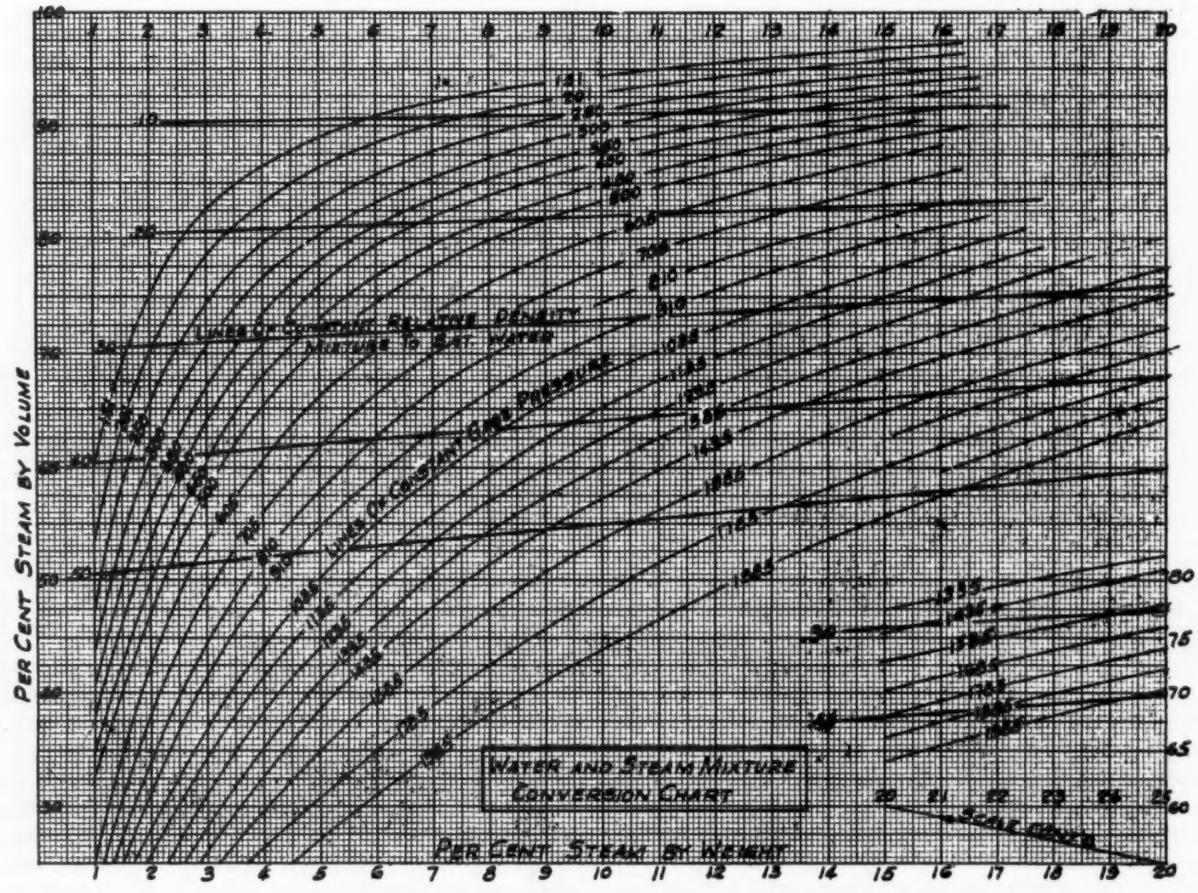


Fig. 10—Curves showing relative densities of steam and water in wall tubes

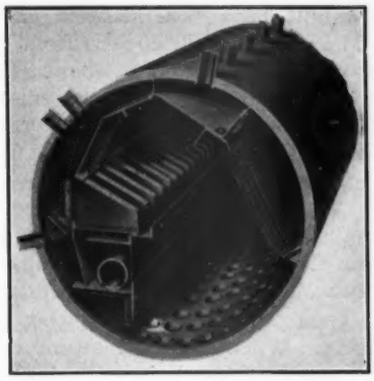


Fig. 11—Bubble type steam washer

and vigorous ignition; and turbulence to insure mixing of oxygen and combustible gas. These three factors will produce a temperature sufficient to melt steel and to make fluid practically any ash. Such temperatures are not necessary to insure complete combustion but are the result of complete combustion. Henry Kreisinger, in a recent paper, covered this subject thoroughly,[1] and R. F. Davis[2] states that "Once ignition has commenced there is a definite advantage in maintaining the flame temperature as low as possible. . . . Providing ignition is not retarded, the maximum cooling surface should be installed and for this purpose bare boiler tubes will give the maximum heat absorption possible." All of this is preliminary to the statement that water-wall furnaces do not retard or impair combustion in any way; in fact, such furnaces may improve combustion.

Where bare tubes constitute the boundary walls of a furnace it is essential that the circulation in those walls be such that there will always be sufficient water in the tubes to insure adequate cooling.

There are little or no data on the friction losses of mixtures of water and steam flowing in the tubes; consequently the proportioning of risers and downcomers must be based on experience coupled with an analysis of probable velocities.

For visualizing the effect of varying percentages of steam and water in wall tubes and the relative densities of such mixtures to the density of water at the corresponding saturated temperature, the curves in Fig. 10 may prove interesting. It takes little head to move water and mixtures of steam and water at low velocities. Friction losses increase as the square of velocity, as do entrance losses; therefore, the velocity of the mixture in risers must be kept within reasonable figures.

It is good practice to avoid any recirculators in water walls; that is, all of the water that enters the bottom of a water wall is discharged in the drum as water and steam. There can be no building up of concentration in these walls.

In both dry bottom and slagging bottom furnaces the water walls are suspended either from the upper wall headers or from lugs welded to tubes at an elevation as close to the upper wall header as is possible. With this design the total expansion of walls and risers is divided and provision for expansion of risers is more easily made.

[1] Paper presented before Joint Meeting of Detroit Section of A.S.M.E. and The Engineering Institute of Canada, November 16, 1937. Published in the May 1938 A.S.M.E. *Transactions* and abstracted in COMBUSTION of January 1938.

[2] R. F. Davis, in a paper before the Institution of Mechanical Engineers (Great Britain) on November 19, 1937 and printed in *Engineering*, London, December 10, 1937.

In the majority of designs for which the author has been responsible the basic boiler design of Fig. 5 has been incorporated, modified to that shown in Fig. 6 or Fig. 7. In such designs all of the water walls discharge into the smaller of the two upper drums, usually called the front drum. Sufficient steam separation takes place in this drum so that little water is carried to the rear or off-take drum. Because of the size and consequent low velocity of steam across the drum most of the entrained moisture is dropped out and steam of better than 99½ per cent quality leaves the boiler.

A further inherent feature of this design is the fact that when the velocity of steam through the circulating tubes is correctly proportioned to the pressure and capacity, and the water level is controlled at the off-take drum, the level in the front drum becomes lower as the capacity rises; therefore, the greater the amount of steam released in the front drum, the greater the volume available for such release.

In low-pressure plants (300 lb and below) there have been relatively few cases where turbines have lost power because of blade deposits. With high-pressure boilers, however, such deposits may be a serious problem. For-

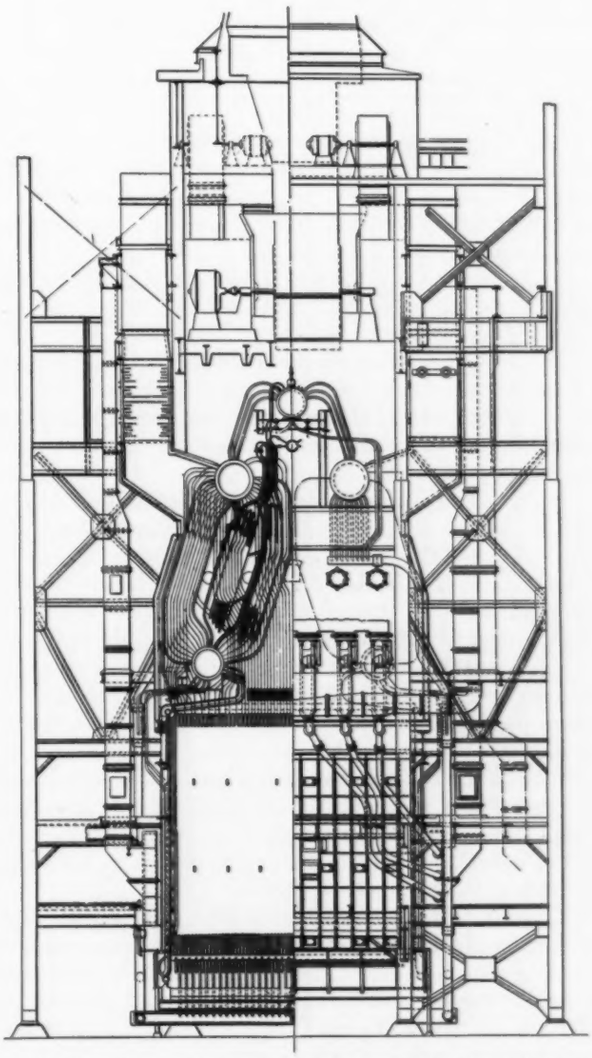


Fig. 12—Five-drum double-set boiler capable of producing a million pounds of steam per hour at 1340 lb pressure and 910 F total steam temperature

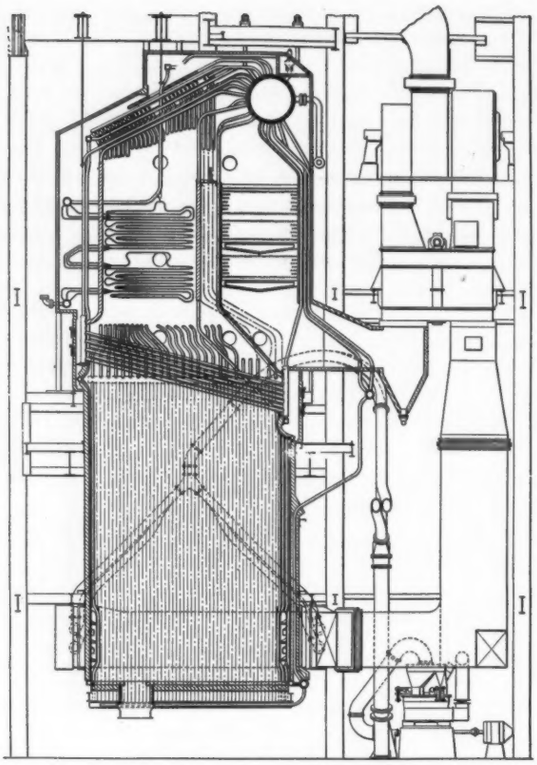


Fig. 13—Section through one of four Waterside units

tunately, methods have been devised whereby the solids normally present in the moisture in steam may be removed or partly removed by the so-called "washing" of the steam with boiler feedwater.

If the boiler water contains 1500 ppm concentration and the steam $^1/_2$ per cent moisture, the steam will then carry $7^1/_2$ ppm of salts.

The theory of steam washing is that the moisture and contained salts mix with the boiler feedwater and the water carried out of the boiler by the steam has a concentration equal to that of the mixture of moisture in the steam and feedwater.

Fig. 11 illustrates a steam washer of the bubble type which has proved very satisfactory. Steam enters the drum through the two rows of circulating tubes located behind the series of hoods. Feedwater enters through the pipe shown in the lower left hand part of the drum, fills the feed trough to a level about $^1/_2$ in., above the row of holes at the bottom edge of the multiple hoods and thence flows over the sawtooth weir into the drum. The steam passes into the hoods and through the small holes up through the feedwater, producing considerable turbulence in the space between the hoods, and a thorough mixing of the steam with the feedwater. The clean steam leaving the hoods passes across the drying screens at the right-hand side of the drum, thence to the steam circulators to the superheater.

A recent test on a washer of this type on the high-pressure boiler at the Nebraska Power Company showed 0.7 ppm with a boiler water concentration of 3020 ppm. A similar washer installed at the Richmond plant of the Virginia Electric & Power Company gave a comparable performance.

Fig. 12 illustrates a five-drum double-set boiler at the River Rouge plant of the Ford Motor Company. Tests of this unit have demonstrated excellent performance. The range in efficiency is from 88 per cent at 400,000 lb of steam per hour to about 84 per cent at 1,000,000 lb. The superheat curve is remarkably flat, the design point being 900 F and the performance range from 810 F at 400,000 lb to about 910 F at 1,000,000 lb. The results of seven tests on this boiler have been published.[3]

Fig. 13 shows one of four boilers installed at Waterside Station of the Consolidated Edison Company of New York, Inc., two of which have been operated since October 1937. Their rated capacity is 500,000 lb per hr at 1350 lb pressure. When operating at any capacity between 200,000 and 500,000 lb per hr slag flows freely and continually. The ash-fusion temperature of the coal used is 2650 F. For the past six months these boilers have had an availability factor of slightly better than 90 per cent.

A unit of almost identical design is installed at the Northeast Station of the Kansas City Power and Light Company, the principal difference being in the capacity and steam temperature requirements, which are, capacity 300,000 lb per hr, steam temperature 815 F, reheat temperature 700 F. The furnace design is based on a specified ash-fusion temperature of 1950 F. From operating data so far obtained this unit has been entirely free from slag on the inclined rows of boiler tubes or on the superheater, although it has been operated at capacities ranging from 250,000 to 325,000 lb per hr. A light spongy accumulation builds up on the refractory between water-wall risers, which usually falls into the furnace. In corners this accumulation may bridge to the

[3] For performance of this unit see COMBUSTION, April 1937.

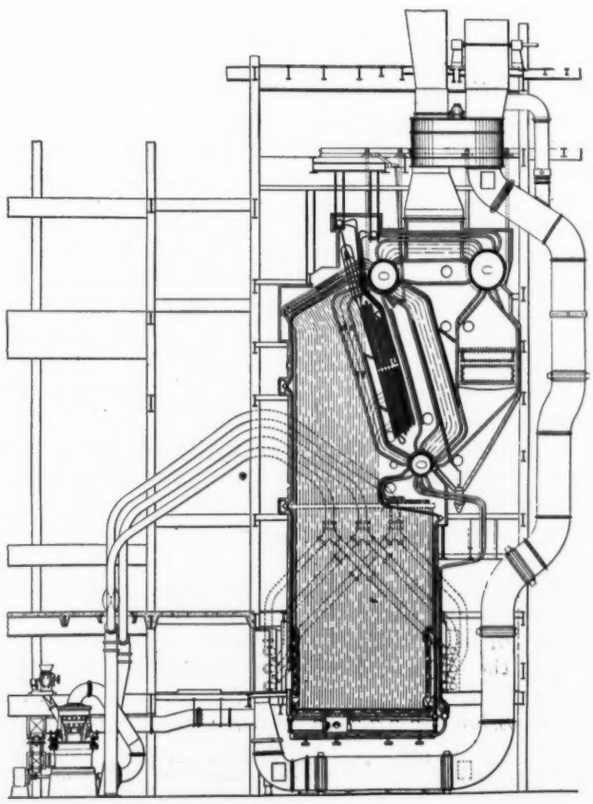


Fig. 14—750,000-lb per hr, 1350-lb pressure unit being erected at Windsor Station

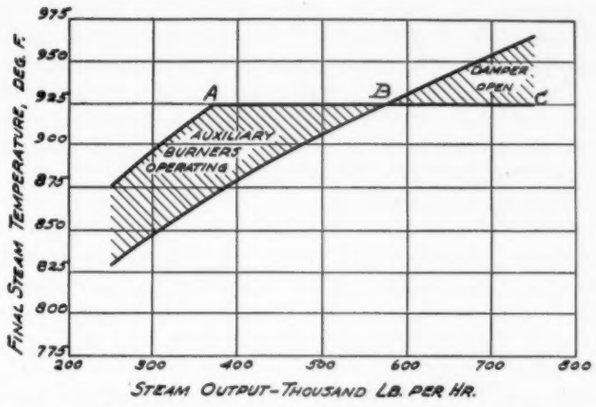


Fig. 15—Representing superheat requirements at Windsor

refractory over the upper water-wall headers and occasionally must be blown off with a lance.

Fig. 14 shows the unit now being erected for the Ohio Power Company at Windsor. This unit is designed for a maximum evaporation of 750,000 lb per hr at 1350 lb pressure. The superheat requirements are shown on the curve, Fig. 15. From point B to C constant steam temperature is obtained by a gas bypass damper. From A to B the temperature is maintained at 925 F by using auxiliary burners located in the upper front wall.

Fig. 16 shows one of two 1400-lb pressure units now being erected at the Schuylkill plant of the Philadelphia Electric Company. The capacity of each is 600,000 lb per hr and 660,000 peak capacity. The steam temperature is 935 F.

A question that frequently arises is, under what conditions is a slagging-bottom furnace preferred over a dry-bottom furnace?

The principal conditions which determine the practicability and desirability of a slagging-bottom furnace are, load factor, fusion temperature of the ash and amount of ash in the coal. Generally speaking, with a low-fusion ash and a high load factor, a slagging bottom is preferable. For ash-fusion temperatures of 2000 to 2250 or 2300 F the load may vary from full to less than one-half load without difficulty in disposal of ash. As the fusion temperature of ash increases the range of load over which satisfactory fusion and flow of ash occurs is smaller. If the ash-fusion temperature is in the 2600 to 2700 F range the load, generally speaking, should be held at or near the 1.0 to 0.75 range.

Formerly most slagging furnaces were of the intermittent or tapping type. Most, if not all, of the more recent installations are of the continuous flow type.

The principal advantage of this type is the fact that there is no pool of molten slag of several inches in depth. If a leak develops in the bottom or sides of an intermittent slagging furnace, the slag in flowing over or on a tube in the bottom or side will cut into the tube wall or burn through the tube. The extent of damage will depend on the weight and temperature of the slag flowing in a given time and the sectional area of the slag stream. For example, in one case a stream of slag of $1/4$ in. diameter, totaling about 25 lb, flowing onto a horizontal tube, cut a hole through the tube wall.

In the designs shown the floor of the furnace is formed by horizontal fin tubes connecting the front and back bottom headers. These tubes are supported at intervals by structural members which are suspended from the headers. On top of these floor tubes is a layer of dry chrome ore about 2 in. deep or three and a half inches above the centerline of the tubes. Below the tubes is refractory backing, insulation and a steel plate casing. Dry chrome ore is preferred to a monolithic chrome bottom as the latter will crack and open with temperature changes.

The side walls of the furnaces are made up of 3-in. tubes spaced on $5^1/_8$-in. centers, the space between the tubes being closed by segmental fins 1 in. wide, welded to the tubes. On slagging furnaces these fins are $1/_2$ in. thick, 1 in. wide and 1 in. long, and are welded by the resistance-weld method. Back of the fin tubes is laid a course of tile which conforms to the shape of the tube and fins; next comes $3^1/_2$ in. of rockwool and a flat-panel steel casing. The outside casing temperature of a furnace of this design is about 120 F.

Because of the flexibility of this type of wall, 30 to 35 ft high, it is necessary to provide round-about buckstays so spaced that the furnace walls will withstand an internal pressure of 20 to 30 lb per sq ft. The wall tubes are tied to these round-about stays to hold the casing and insulation tight against the tubes.

Burners for pulverized coal are of several types usually known as horizontal turbulent, vertical turbulent, vertical and horizontal intertube, Lopulco vertical fantail and tangential.

Tangential burners produce the greatest degree of turbulence and may be used with either dry-bottom or

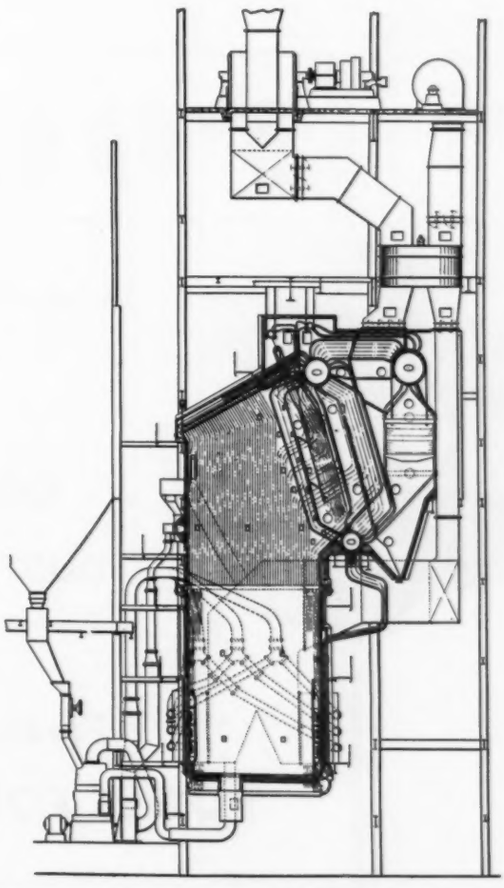


Fig. 16—One of two 1400-lb, 600,000-lb per hr units for Schuylkill Station in Philadelphia

slagging-bottom furnaces. Tangential burners have been designed to use oil, natural gas and blast furnace gas, either singly or simultaneously and in any proportions. These burners may also be adapted to burn coal, oil and natural gas or coal, blast furnace gas and oil.

Horizontal turbulent burners are easily arranged to burn coal, oil and natural gas.

Because of the higher rate of heat absorption obtained, tangential firing is preferred for large units. Distribution of heat to the walls is more uniform than with other methods of firing and furnace exit temperatures lower.

The usual specification for such boilers calls for a complete steel casing for the entire unit. It is my opinion that such casings for the furnace walls above the side and front wall headers are unnecessary and have little value except as to appearance. They are expensive because of the number of access, inspection and lancing doors and soot blower inserts which necessitate irregular or odd shaped panels. Cracks in brickwork cannot be easily detected and it is very difficult to trace leaks.

For upper walls of moderate height a solid brick wall with insulation properly tied to buckstays will cost considerably less than a cased wall.

Large single-drum and three-drum boilers are preferably suspended by U-bolts from overhead. This method of suspension is superior to the use of brackets attached to the underside of drums resting on steel girders below the drums. The U-bolt suspension is more flexible and permits free expansion of the drums.

Where drums rest on girders, unless rollers are used between brackets and girders, the stress parallel to the flanges may be excessive. Such girders must be ventilated or insulated; in either case the continuity of the upper side-wall brickwork is broken and the walls are weakened.

The rolling of heavy wall tubes requires considerable care and understanding of the various factors that enter into the problem. It is our practice to bore the ends of heavy wall tubes to insure concentric tube ends and a uniform wall thickness.

Tubes should be soft, not over 66–67 Rockwell B hardness. Narrow tube seats are preferable to wide seats, but in thick drums counterboring to make narrow tube seats requires thicker drums. Therefore, a compromise is made and tube seats as wide as 2 to $2^1/_2$ in. are used. Wide seats require more rolling. When tube holes are counterbored care is taken that on the lower drum the tube seats on the upper half are counterbored from the inside so that there is no annular space between tube and drum to collect ash and moisture condensed on the tube when the boiler is cold. Such pockets will collect acid, and the tubes and drum will corrode. On upper drums the counterbore may be on the outside, less metal is removed and the drum may be thinner than where counterbored on the inside.

Such operating data as are available at this time lead to the following conclusions: that the designs presented, which comprise the application of water-cooled walls to boiler types, and which have had a satisfactory performance record, have been along sound engineering lines; and that they will meet the demands of those requiring dependable, efficient and economic high-pressure steam generating units of large capacity.

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The Ramsin Once-Through Boiler

By T. SAUER
Berlin

This description, taken from *Die Wärme* of January 15, 1938, deals with a boiler of 440,000 lb per hr maximum capacity at 498 lb steam pressure which is one of a number of Ramsin boilers of different pressures which are being built by the Russian Government in conformance with its industrial plan. Of particular interest is the table containing design data, including steam and water velocities and the hydraulic friction through the tubes.

FAVORABLE operating results obtained with the first Ramsin boiler of 440,000 lb per hr capacity and 1990 lb steam pressure, and the relative ease with which such boilers of standardized design, without drums, can be built in USSR, led the Commissariat for Heavy Industry to undertake the construction of a series of such units. Of several, designed for different operating conditions, the 440,000-lb per hr, 498-lb pressure boiler will be described. This has the following general specifications:

Maximum steam output,	440,000 lb per hr
Normal steam output,	352,000 lb per hr
Steam pressure,	498 lb abs
Steam temperature,	806 F
Feedwater temperature,	302 F

The feedwater, consisting of pure condensate, after being heated in regenerative heaters to 302 F, enters the 6420-sq ft economizer and, at normal output leaves at 437 F and a pressure of 710 lb abs. This water is led from the upper header of the economizer to the lower header of the radiant heating surface and then distributes into forty parallel circuits of carbon steel tubes varying in diameter from 1.77 to 1.97 in. The first portion of these circuits is located in the lower third of the furnace which is hopper shaped. The coils are inclined and arranged in spiral form but rectangular in plan. The arrangement is here shown diagrammatically and the parallel circuits are represented by a single line. The circuits extend in this way to the upper part of the furnace. The normal vertical spacing between the tubes is about $2^1/_2$ in. in the lower part of the furnace and somewhat greater in the upper portion.

In the first gas pass of the top (see illustration) the circuits change to four horizontal rows of ten tubes each, thence into two rows of twenty tubes each at the entrance to the convection section of the unit. All forty tubes then turn back, continue along one side of the furnace to the front and then turn up to form the roof of the furnace and enter the steam header. They are here spaced 2.6 in. apart. From the header the steam passes to the convection superheater.

The coils in the lower portion of the furnace form the radiant water preheating section, the remaining portion being considered as the steaming section.

The total wall heating surface is 11,300 sq ft, of which the water preheating section accounts for 688 sq ft; that in the first pass is 2100 sq ft and the roof surface 366 sq ft, making a total of 13,766 sq ft. The superheater surface is 7330 sq ft and consists of 182 tubes of $1^1/_4$ to $1^1/_2$ in. diameter.

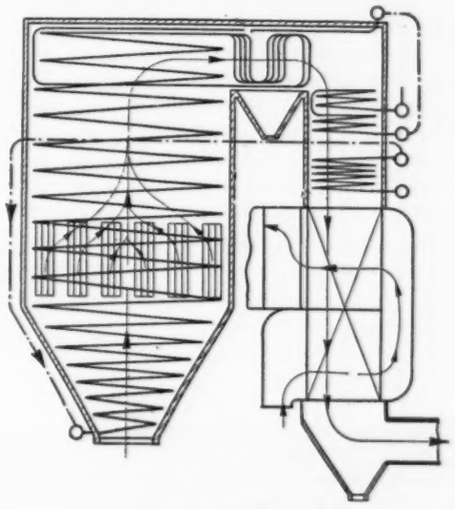


Diagram representing circuits in Ramsin boiler

The air heater which follows the economizer is of the vertical tubular type, having 55,400 sq ft of surface and consists of 4368 two-inch tubes. The gas velocity

DESIGN DATA FOR 440,000-LB PER HR, 498-LB PRESSURE RAMSIN BOILER

	Furnace	First Pass	Superheater	Water Preheater	Air heater
Nominal heating surface, sq ft	3600	2470	5980	6420	55,400
Temperature of the gases leaving the heating surface, F	1922	1600	1120	716	392
Temperatures of water, steam and air leaving the heating surface, F	482	518	806	437	554
Useful heat absorption, Btu/hr	244.5	46.6	62.0	51.6	56.0
Heat absorption, Btu/sq ft/hr	67.9	18.9	10.35	8.0	1.01
Heat transfer coefficient, Btu/sq ft/ deg F/hr	...	13.3	14.75	14.75	4.1
Average gas velocity, ft/sec	15.75	51.9	60.5	53.8	50.6
Draft loss for gas, in. of water	...	0.79	2.1	2.1	1.3
Average air velocity, ft/sec	...	...	...	...	2.33
Draft loss for air, in. of water	...	...	...	...	3.46

	RADIANT HEATING SURFACE Furnace	Boiler	Superheater			
Average velocity of steam and water within the tubes, ft/sec	2.84	61.4	129.0	75.8	2.13	....
Hydraulic resistance, lb/sq in.	0.284	107.3	30.4	27.6	2.85	....
		138 lb/sq in.				

Total gas resistance in the boiler amounts to 6.26 in. of water
Total water resistance in the boiler amounts to 168.5 lb/sq in.
The calculated efficiency amounts to 88 per cent.

through the heater is 44.2 ft per sec and the air velocity 24.6 ft per sec.

The furnace is designed for burning pulverized coal or oil, either separately or in combination and there are 12 burners, six located on each of two opposite sides of the furnace. Automatic combustion control is provided as well as remote manual control.

For the boilers designed to operate at moderate pressures centrifugal boiler feed pumps are employed and the proper ratio of water to fuel for different loads is obtained by means of proportioning valves.

Units designed for high pressures are provided with feed pumps of the plunger type and the correct ratio of fuel to water is accomplished through driving the coal feeder by means of an oil-operated fluid motor. The oil for driving this motor is supplied by the same pump that serves the oil burners. The fuel-water ratio control is on this pump so that for a given load the same amount of oil is pumped whether it be supplied direct to the burners or used in operating the fluid motor when burning pulverized coal. Of course, in the latter case the oil, after driving the fluid motor, is returned to the sump.

The overall dimensions of the unit described are—height 47 ft, width 28½ ft and length 47 ft. Design data are given in the accompanying table.

The high-pressure units are similar to this boiler but require more costly construction. A unit of 110,000 lb per hr capacity designed to operate at 2000 lb pressure is now being built.

On Power Show Committee

Nine new members have been appointed to the Advisory Committee of the Thirteenth Exposition of Power and Mechanical Engineering, which will be held at Grand Central Palace, New York, during the week of December 5 to 10. The new members of the Committee are Comfort A. Adams, E. G. Budd Manufacturing Company; J. T. Barron, Vice-President, Public Service Electric & Gas Company; R. E. Dillon, Vice-President, Boston Edison Company; Crosby Field, President, American Society of Refrigerating Engineers; A. C. Fieldner, Chief Mechanical Engineer, U. S. Bureau of Mines; N. E. Funk, Vice-President, Philadelphia Electric Company; George L. Knight, Vice-President, Brooklyn Edison Company, Inc.; Joseph Pope, Stone & Webster Engineering Corporation; and Alfred Vaksdal, Plant Engineer, Corning Glass Works.

Publicity on Coal Costs

John Carson, Consumers' Counsel, has gone into Court with a demand for full publicity on all factors involved in fixing minimum prices for bituminous coal.

In a brief supporting the Coal Commission's ruling that cost reports of individual producers may be introduced in the hearings, Mr. Carson argued that "the secrecy attending private, competitive price fixing is incompatible with the full disclosure and full public record upon which public action must be predicated."

A MONEY-SAVING HINT ON INSULATION by "Springy Ball"

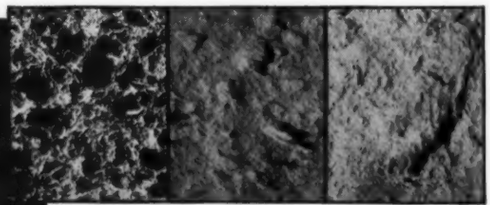


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As reported in the foreign technical press

Grooved Boiler Tubes

According to an article by R. Schulze in *Die Wärme* of March 12, a new type of grooved boiler tube has been developed by the Mannesmannröhren-Werke in Germany. A spiral groove is cold-rolled on the outside of a seamless tube to a depth of about $^3/_{16}$ of an inch and about $^1/_4$ in. pitch. The tube is then heat treated. The heat transmission, as indicated by tests quoted, is about 30 per cent greater than that of a plain tube of equal diameter because of the greater exposed surface and it is claimed that the mechanical qualities are equal to those of the plain tubes. The grooved tubes can be bent, expanded or welded in the same manner as plain tubes and are being made for boilers, superheaters and economizers, particularly those of the steaming type.

Small Radial Turbine

Siemens-Schuckert Werke has developed a radial-flow back-pressure extraction turbine of the single-rotation type for high steam pressures and relatively small steam flow. This is described by O. Schöne and C. Krafft in *Zeitschrift des Vereines deutscher Ingenieure* of April 16.

Such a machine has been operating at the cable works of that company in Berlin for the past two years. It is of the double-casing gearless type with the high- and low-pressure sections at opposite ends of the generator and has a maximum rating of 5000 kva at 3000 rpm. Operation is at variable pressure of 1780 to 710 lb per sq in., according to the steam requirements, the back pressure being from 150 to 100 lb and the extraction pressure from 680 to 400 lb. There is a total of 81 stages, the diameter having been kept to a minimum and the length of blades increased to reduce leakage. A thermal efficiency of 70 to 72 per cent is claimed for the unit.

Pulverized Coal in Australia

Within the last four years the capacity of pulverized-coal-fired boilers in Australia has doubled, according to a chart in the May issue of *Engineering and Boiler House Review* (London). The first installation was made in 1922. The grindability of coals from the southern fields of New South Wales compares favorably with that of some of the softest coals in the United States, but the coals from the western fields are much harder. Most Australian coals have a relatively high ash fusion temperature, ranging from 2250 to 2900 F.

Some time ago the State Electricity Commission of Victoria erected an experimental plant for burning brown coal in pulverized form, but this did not prove attractive and all subsequent installations burning brown coal have employed stokers.

The article contains a tabulation giving details of nineteen pulverized-coal-fired plants, built or under construction. The largest and most important of these are the Bunnerang Extension and the East Perth Station. The former, scheduled for initial operation in 1939, will contain four 350,000-lb per hr boilers producing steam at 650 lb, 850 F; and the latter, which is expected to go in service next October, will contain three 135,000-lb per hr units operating at 650 lb, 830 F.

Tangential Firing

The March 19 issue of *Génie Civil* contains an article by P. Jarrier discussing tangential corner firing. Several German installations are cited in which greatly increased capacity was obtained by substituting this type of firing. Among these was the Böhlen central station where the capacity of one of the boiler units was increased from 160,000 to 330,000 lb of steam per hour by replacing the vertical burners with tangential corner firing and increasing the water-wall surface. Another unit in the same plant was raised in capacity from 160,000 to 400,000 lb of steam per hour by similar means. Mention is made of the Mannheim installation whose boilers employ tangential firing in producing 280,000 lb of steam per hour at 1540 lb per sq in. with very low volatile coal—only 7 to 8 per cent.

Gas Producers for Marine Use

According to *Engineering* (London) of May 27, Germany, in an effort to reduce dependence upon imported liquid fuel, has embarked upon the introduction of the gas producer for propulsive power of ships. Already twenty producer gas-driven ships have been built or are building in Germany. These producers operate with coke, anthracite or wood and are said to get results comparable with Diesel drive. To date the gas producer has been confined to relatively small ships of the river class requiring 750 bhp or less. The fuel consumption per bhp when using anthracite ranges as low as 0.92 lb per bhp.

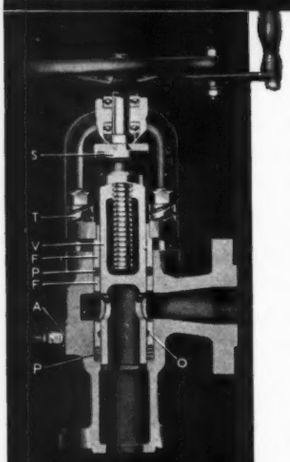
Rebuilt Versus New Turbine

In an article in *Die Wärme*, Dipl.-Ing. Hellmut Eickemeyer discusses the rehabilitation of old, existing or second-hand equipment in preference to new units as one means of meeting the shortage of certain materials and skilled labor in the metal industry and in line with the present German four-year plan.

Despite the necessity for considerable reconstruction and renewal of parts in many cases, the author contends that the old unit frequently figures out to be the cheaper, in view of present labor and material costs in Germany. The technical requirements for re-use must,

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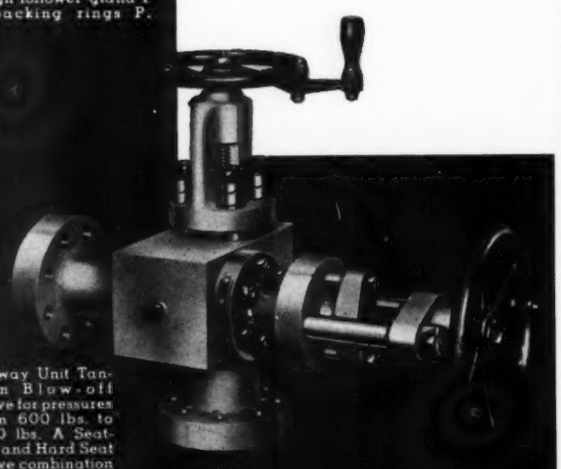


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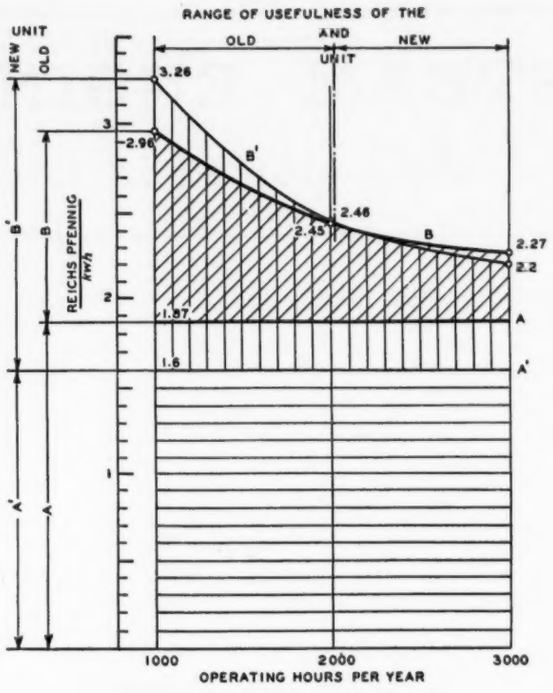
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Comparison of Operating Costs

AA' = cost of heat; BB' = (1) charges for interest and depreciation, (2) power consumption of condenser, (3) supplementary oil use.

of course, agree in some measure with the known specifications of the available unit and, aside from investment costs, including reconstruction, anticipated operating results are important. He cautions against under-estimating fixed charges which must take into account the probable shorter life of the reconstructed unit.

The author cites, as an example, the reclaiming of a 4000-kw condensing turbine, built in 1914, for recent use in a chemical plant. This plant formerly operated on hydro power, the surplus of which was used to generate steam in an electric boiler to supplement that supplied by low-pressure steam boilers for process. With increased plant output both the steam and power requirements rose rapidly and it became necessary to install a 600-lb boiler supplying a back-pressure turbine which exhausted to process. Subsequently, this unit fell short of meeting further power demands. These could have been just about met by increasing the boiler pressure, but this was not considered inasmuch as there would still have been a deficiency during the winter months when there was a shortage of hydro power. Because of the relatively short duration of the winter peak it was therefore decided to carry it on a condensing unit to be operated with steam from the old 200-lb boilers.

A study of the probable load requirements during the period of minimum hydro supply indicated that a 4000-kw condensing unit would be required. A used unit of such capacity was investigated and it was found that much of the steam end would have to be replaced, the generator rewound and a new condenser provided, but that the total cost would be less than that of a complete new unit. Comparison was made of fixed charges as between the rebuilt machine, for a given life and probable operating hours, with those on a new machine. The results showed (see curve) the operating costs to be favorable to the old machine for operation up to 2000 hr per year. As the water shortage ranged from 1000 to 1500 hr per year the decision was in favor of the used turbine.

REVIEW OF NEW BOOKS

Any of the books reviewed on this page may be secured from Combustion Publishing Company, Inc., 200 Madison Ave., New York

Elements of Steam and Gas Power Engineering

By A. A. Potter and J. P. Calderwood

This is the fourth edition, the text of the 1930 edition having been largely rewritten and brought in line with developments in practice during the interim. Also, most of the illustrations dealing with equipment and its use are new.

The text is divided into three parts, the first dealing with steam power in its various aspects, the second with internal-combustion engines and the third with the application of steam and gas to motive power. Although written primarily as a textbook for engineering students, the treatment is non-technical and provides a comprehensive review of current practice. Of particular interest is the first chapter dealing with the social significance of power including pertinent statistics and the relative importance of different types of prime movers.

There are 374 pages, 8 × 5½ in., fully illustrated, with cloth binding; price $2.75

Combustion Third Edition

Prepared by American Gas Association

This is a 200-page handbook, 9 × 12 in. size, containing engineering data and basic information on the chemistry of combustion, heat transfer and materials entering into combustion where commercially available gases are employed.

The chapter on Thermal Capacity is replete with data on the specific heat and the thermal properties of various metals, liquids and gases, and that on Heat Transfer discusses conduction, convection, radiation and the heat transfer through insulation. Types of gas burners are described, air-gas ratios are discussed at length and various methods of temperature control dealt with. At the end of each chapter there is appended a full bibliography of published information for convenient reference.

The subject is handled in a thoroughly practical manner and the numerous charts and tables should prove useful to the engineer concerned with gas burning problems. The price of the book is $2.

Steels for the User

By R. T. Rolfe

The aim of this book can perhaps best be expressed by quoting from its preface in which the author observes:

"The engineer and user, who is always seeking further knowledge about his materials, must pick his way through a multitude of volumes in the endeavor to construct for his own use a satisfactory fabric of knowledge which will enable him to exercise a greater discrimination and to select with more assurance the steels for different purposes.... Existing books are usually written by scientific men who are less conversant with practice.... The present work constitutes an attempt to bridge this gap between science and practice for carbon steels in industry."

The text covers chapters on Mechanical Quality and Its Assessment; Specified Requirements for Commercial Steels; The Effect of Composition upon Mechanical Quality; Bright and Free-Cutting Steels; The Heat Treatment of Steel, both high and low carbon; Case-Hardening and Similar Treatments; Use of Steels at Elevated Temperatures; Fatigue Testing; and General Principles of Selection. Many illustrations, curves and test data are included to supplement the text.

The book is a practical one, scientifically explained, rather than a theoretical one with practical examples, and the author, who is chief metallurgist with W. H. Allen, Sons & Co. Ltd., Bedford, England, appears to have accomplished his purpose in a very creditable manner.

There are 280 pages, 8½ × 5½ in.; price $8.50.

Mechanical Engineers Handbook—Design and Shop Practice

By Robert T. Kent, Editor-in-Chief

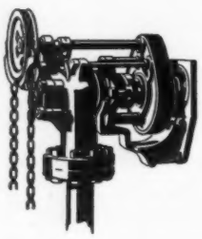
The present eleventh edition of Kent's Mechanical Engineers Handbook is issued in two volumes, that dealing with "Power" having been brought out in 1937 and that on "Design and Shop Practice" in 1938. In this way it has been possible to devote greater space and more thorough treatment to the various subjects covered without making the book too bulky for convenient use; also, inasmuch as this division corresponds with the two general occupational classifications of mechanical engineers, the needs of many will be met by one or the other of the volumes.

The text for the present volume on "Design and Shop Practice" has been largely rewritten and covers the General Properties of Materials, Iron and Steel, Corrosion and Corrosion-Resistant Metals, Non-Ferrous Metals and Alloys, Non-Metallic Materials, Fabricated Materials, Strength of Materials, Mechanism and Mechanics, Fastenings, Mechanical Springs, Rotating Members, Bearings and Lubricants, Gearing, Control Mechanisms, Vibration and Noise, Structures and Buildings, Industrial Heating Processes, Forging Practice, Foundry Practice, The Machine Shop, Materials Handling, Mechanical Power Transmission, Electric Motors and Their Control, Miscellaneous Shop Equipment, Safety Engineering and Mathematical Tables.

The price is $5.

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